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HINTS ON THE INSPECTION OF STEEL.

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It has probably been the experience of all inspectors of engineering material that, upon being assigned to duty in connection with such work, their labor has been very severe at first on account of the lack of definite information as to what was required of such officers.

In presenting these notes it is hoped that the information so long desired by many inspectors may in part be furnished. As it is an exceedingly difficult matter for any one to draw up specifications that will be absolutely clear and complete, the hints and suggestions herein contained may be of value to many employed on such duty.

In the inspection of Government material there are practically three parties who are interested in construing the specifications: The Executive Department of the Government which requires the material, the Contractor who furnishes it, and the Inspector, who is presumed to be sufficiently expert to cope with the well-paid and highly-trained specialist retained by the manufacturer

or contractor. Something is needed to bring these three interested parties to a more rational understanding, and it is hoped that these "hints" will not only in part effect this purpose, but also have a tendency to expedite business.

The hints are the results of experience, and represent careful and conscientious work. In their preparation all available sources of information have been drawn upon, and it is only because some such brochure is a necessity to efficient naval inspection that the task of compiling the experience and observation of many people was undertaken.

The general hints given under Section IV relating to Structural Steel will likewise apply to the sections pertaining to Steel Plates, Steel Forgings and Steel Castings.

I.—STEEL PLATES.

The size of ingots used for steel plates varies with the mill in which the plate is to be rolled, and with requirements of the specifications. In the case of large plates, some mills prefer to cast separate ingots for each plate, the ingot being cast in such a shape as to be most conveniently rolled to the required dimensions. The practice in other mills is to cast large ingots, which are to be rolled in the slabbing mill to a certain width and gauge, and cut into suitable lengths for the plates required. On account of excessive segregation in large ingots, it is undesirable to have more than eight tons in an ingot. Where plates are rolled direct from small ingots, the inspector checks the heat numbers on the ingots and on the finished plates, there being not many intermediate stages in the process of manufacture for him to follow. In the case of large ingots from which a number of plates are rolled, the following method of keeping track of the steel in the mills has proved satisfactory:

METHOD OF NUMBERING SLABS.

The numbering usually commences with "1" at the first of the year. Each slab as it is weighed at the shears is stamped with the consecutive number until the end of the year. This number is called the *slab* number.

In addition to the number that is stamped on a slab, as above described, each *slab* is numbered in accordance with its position in the ingot. The first cut from the top of an ingot is No. 1, the second No. 2, and so on to the last cut at the bottom of the ingot. In shearing the rolled-down ingot into slabs, the top of the ingot is not necessarily sheared first. Sometimes the shears begin with the bottom, or after an ingot is partly sheared from one end it may be reversed and sheared from the other. In whatever way it is sheared or the slabs numbered, the numbering of the individual cuts begins with 1 at the top and runs consecutively to the bottom. These numbers are called "cut numbers," and are recorded by the weigher on the sheet and sent to the plate mill with the ingot numbers, heat numbers, slab numbers, and weights of slabs. Each slab should have stamped upon it the heat and slab number, by means of which the exact history of the slab can be traced. The inspector can procure these numbers in the plate-mill office either from the slab-mill sheet or from the record book in which they are entered. It is important to know from what part of an ingot a plate is rolled, in order to test for uniformity of material due to segregation.

GENERAL PROCEDURE OF ROLLING AND MAKING REGULAR TESTS.

Two ingots are rolled into slabs; then four slabs, two from each ingot somewhere near the center cut, are selected by the mill and rolled into plates. From these first four plates rolled the samples for tensile tests are cut. If they pass the tests two more plates are rolled for bending tests, and, if they pass these, the heat is accepted, subject to any special test that the inspector may see fit to make.

When the ordinary charge of six slabs is placed in the heating furnace to be reheated for rolling into plates, a heat sheet is made out in the plate-mill office, giving the number of heat, order number, dimensions and gauge, and location marks of each plate to be rolled, a copy of which is given to the inspector. If coupons for tests are to be taken from the plates they are indicated on this sheet.

SPECIAL TESTS.

From time to time such additional tests may be made as may be deemed necessary to detect lack of uniformity in the material.

If the tensile strength is near the upper limit, the inspector should not allow thinner plate than the test plate to be rolled without making special tests; or, if the tensile strength is near the bottom limit, he should not allow thicker plate to be rolled without special tests. He may, however, when the tensile strength is high and the elongation good, allow thicker plate to be rolled, or, if tensile strength is low, allow thinner plate to be rolled. Plates from ten pounds to fifteen pounds per square foot, or from fifteen to twenty-two and a half pounds, or from twenty to thirty pounds, if the conditions are favorable, will come within the specified limits. It is well, however, for the inspector to make special tests to satisfy himself as to what certain steel will do.

Unsatisfactory Regular Tests.—Sometimes a heat will pass the prescribed tests of pulling and bending and yet not give entirely satisfactory results. On a heat of this kind, a good many special tests should be made.

Heating and Rolling.—It is well to watch the rolling as closely as possible to see that all plates receive as careful attention as the plates rolled for tests. It is hard to lay down any rule for finishing temperature and final passes, the effects of which the inspector can determine by special tests. If a test plate is designedly finished at a very low or very high temperature, and other plates from the same heat are finished at temperatures differing greatly from the test plates, special tests should be made.

Sometimes a plate is rolled transversely to the length of the ingot, but generally it is rolled longitudinally. The results from these two methods seem practically the same. The mill will often, however, roll transversely when transverse tests are to be taken. It would accord better with the specifications if all test plates were rolled in the same direction as the remainder of the heat.

If long and narrow plates are to be rolled from heats tested in broad plates, transverse special tests should be made.

Tests for Segregation and Uniformity of Steel.—If the top cut (cut No. 1) is rolled into Government steel, always take a coupon for special test. Such a test will generally reject this plate, hence it is the practice of the mill to put this top cut into trade. It is well occasionally to make a special test on the second and bottom cuts. On plates from ingots not tested by regular test one special test should be made.

BURNT PLATES.

If a plate is burnt on the edge or end a special test should be made in such a way as to represent the part nearest the burn. If the inspector suspects a plate of being overheated he should make a special test.

SURFACE DEFECTS.

As a guide for the detection of surface defects the following list is arranged according to importance.

(a) *Pipes* occur in the top and center parts of an ingot, and therefore are to be looked for especially in top cuts. The steel about a pipe cavity is very susceptible to heat and is often found burned; therefore, on top cuts burned ends may indicate inferior steel.

(b) *Cracks* caused by burning or red-shortness. Cracks from burning are likely to be found in any slab owing to its having been burnt in the heating furnace. Cracks due to red-shortness are rare. Steel of this kind would hardly stand test, but in heats already passed it may occur owing to excessive segregation.

(c) *Snakes* occur generally in the bottom half of an ingot, but seldom in the bottom cut. They are the cause of rejection even if they are very small, for their depth cannot be ascertained and they may be the cause of a plate failing in use.

(d) *Blisters, Laminations, and Blow-Holes.*—Blisters show in raised places on the surface of the plate, and their extent can be determined by sounding with the hammer. They, however, taper off into laminations, therefore a plate showing any blisters should be rejected.

Laminations are to be looked for on the edge of the plate after shearing. It is a flaw hard to detect, but often develops in the bending test.

Blow-Holes of perceptible size are of rare occurrence in ordinary steel plates, but sometimes they can be seen on the surface of a plate with their sides pressed together in a distorted shape.

Note.—In examining a plate for snakes and blisters it should be viewed from various directions, since in certain lights these defects cannot be seen.

Blisters, Laminations, Piping, Burns, and Snakes are always cause for rejection. The effect of other defects is left to the judgment of the inspector.

(e) *Rolled or Folded-in Steel.*—A defect in rolling, or due to a very hard spot being rolled into the softer steel.

(f) *Cinder* is generally due to some defect in the heating furnace such as to cause cinders or pieces of fire-brick to be rolled into the metal. Steel in the immediate vicinity of a cinder is likely to be burned and porous.

(g) *Scale*, either red oxide of iron or flakes of steel, like slivers, that may be knocked off with the hammer, leaving a pit with smooth and solid bottom. This pit is generally cup-shaped and should not be mistaken for cinder spot.

(h) *Foreign Substances*, dropped on the steel plate while being rolled and pressed into the finished material.

(i) *Brick.*—Red brick color, generally shallow but often of considerable extent, caused by the heated slab taking up dirt from the bottom of the furnace; therefore, it occurs more frequently on the bottom of plates.

(j) *Pits* made on the surface of the plate by scale or foreign substances.

The importance of a careful surface inspection varies with the use for which the plate is intended. A boiler plate must be practically free from surface defects.

Scale, Cinder, and Brick can be easily removed with the hammer and the condition of the metal in the vicinity be readily determined. In examining a plate for cinder and scale pits the thickness of the plate must be taken into consideration. But even in the thickest boiler plate, if it is to pass muster, these pits must be very shallow and show solid bottoms. While one pit,

cinder, or scale mark may not be sufficient for rejecting a plate, several of the same kind may be.

The plates next in importance to boiler plates are the outside hull plates for vessels.

Thin plates (10 pounds to 12½ pounds) should be as free as possible from surface defects.

In light plates some buckling is permissible. If the buckles run from side to side of plate it can be easily straightened and should not be rejected. If the center part of the plate is buckled and the edges are rolled flat, the plate should be rejected. If the center part is rolled flat and the edges are buckled, the inspector should use his judgment as to the seriousness of this defect. If a plate is to be used as the web of a girder, it should be free from buckles, but if it is to be curved or bent, as is often the case, buckles on the edges are not of so much importance.

II.—STEEL FORGINGS.

Many of the notes following under the head "Structural Steel" apply equally well to the subject of steel forgings. The hints, however, under this head, apply particularly to steel forgings.

In the manufacture of steel forgings the methods employed in heating, forging, and annealing the steel are of the greatest importance. The determination of the character of the forging by physical tests gives merely the condition of the steel in the locality from which the test coupon was taken, and does not expose the defects due to careless heating, working, and annealing.

Small forgings are of necessity heated and worked practically uniformly, and the following of the process of manufacture is not of so much importance, but in regard to large forgings the case is different, and the proper heating, working, and annealing to insure the best results are hard to obtain.

PROCESS OF MANUFACTURE.

Forging.—In the forging of rounds six inches in diameter and over, it is best to use a V-shaped die, as experience has shown that this shape gives a more compact product than the flat die.

The hammering should be worked from end to end of the forg-

ing, but in some kinds of work it is necessary to forge the middle first; there is no objection to this if the subsequent heating overlaps the previous work sufficiently to remove strains, and also if the entire piece is annealed as a final treatment.

It is not good practice to rush a forging through the hammer and then give another pass to smooth it down. Bumps left on a forging, if more than two inches high and with ends at a less angle than 45 degrees to the normal (sectional) plane of the forging, are very objectionable.

No work should be done on steel when it is at the blue heat (about 600 degrees C.), for at this heat the steel is very brittle and likely to crack under the hammer. While these cracks may not be apparent in large forgings, it is very common to see steel plates break in two while being worked at this temperature. This same steel could have been worked with perfect safety when cold or when at a red heat.

Heating.—The heating of metal to be forged is often carelessly done even in the best shops. One end of a bar of cold steel will often be put into a furnace having a temperature of over 800 degrees C.; the other end of the bar being outside of the furnace door. The rapid rise of temperature of the end in the furnace is apt to form incipient fractures in the part of the bar at the furnace door. The higher the percentage of carbon in the steel, the more likely it is to be injured in this way. That cracks, which have caused large forgings to break in two, are thus formed has often been demonstrated in practice, and failures which have occurred after the forging has been in use are attributable to this cause. This difficulty can be overcome by either heating the whole bar to a blue heat before the end is put into the furnace, or by allowing the furnace to cool down and then bringing up the temperature after the end of the bar is inserted.

All forging, the working of which has required more than one heating, or in which the metal has been only partially heated, such as in the case of eye-bars, bent plates, and shapes, should be entirely annealed.

Annealing (particularly eye-bars).—In the process of forging

eye-bars, the head of the bar is subjected to great heat while the body of the bar is not heated at all; consequently, after forging, the steel in the bar is composed of metal varying in hardness and grain, and full of internal strains. To correct this, the bars are placed in an annealing furnace, where they are heated to a temperature of from 650 to 800 degrees C., depending on whether a high- or low-carbon steel is being treated. This annealing temperature is a critical temperature for steel, and is lower for high-carbon than for low-carbon steel. At this heat, the steel is of a medium orange or dull yellow color. It should be held at this temperature until it is uniformly heated throughout its mass, and then allowed to cool slowly. The cooling is generally accomplished by shutting off the heat from the furnace and allowing the furnace and bars to cool together.

Proper annealing relieves internal strains and makes the steel homogeneous. Any increase of temperature over the heat required to give the medium orange color, or holding the steel at this temperature longer than necessary to give it a uniform temperature throughout the body of the bar, may be a cause of injury, as the steel is likely to be burnt, or decarbonized to a greater or less depth. The flame of the furnace should not be allowed to come in contact with the metal. In some shops the bars are placed in the annealing furnace on edge, and the flame enters the furnace through jets placed at intervals about four feet apart along the bottom. The effect these flames have on the metal is seen in testing full-sized eye-bars; for it generally develops that at the points where the flame came in contact with the bar the elongation is greater than elsewhere, excepting the point of fracture. This shows unequal heating and softening of the metal in places, due to defective design of annealing furnace.

In order to anneal properly there is required a furnace of such design that the flame cannot come in contact with the metal nor allow it to become hotter in one part than in another. The experience and judgment of the annealer also play an important part; for, on account of the varying sizes, thicknesses, and carbon contents of pieces to be annealed, he has to vary his temperatures and time of annealing to suit.

III.—STEEL CASTINGS.

Specifications should be well studied and briefed. The inspector should keep himself posted on the analyses of the various heats, and use the means provided to check the work of the contractor's chemist, at least until the accuracy of his work is assured. The sample used for checking should always be identical with that used by the contractor; otherwise, on account of segregation there may be considerable difference in results, although the process used by each chemist may be the same. Drillings from the end of the test bar can be obtained at the time of testing, to supply all wants. These drillings should go deep enough to reach the center of the bar.

Coupons for Testing.—The coupon cast for testing purposes should be an integral part of the casting and should be large enough to form three bars, each one inch square and about six inches long; the test bars should be cut from the coupon singly as needed, each bar being stamped in the presence of the inspector with his identification mark; the heat number, name, and number of casting should also be stamped on it. Should a test bar fail, owing to a flaw showing, another bar might be taken from the same coupon and immediately tested; but should there be no flaw shown, a second test should not be made, in case of failure of the first bar, until the casting has been re-treated by annealing, or otherwise.

The Government specifications call for a test specimen 0.505 inch diameter, which gives very nearly an area of one-fifth of one square inch. (The threaded ends are larger.) But it sometimes happens that a flaw develops in the bar in turning, and when finished the diameter of the bar is less than the 0.505 inch required. The required results can be figured out for the reduced diameter, but the custom of furnishing specimens at diameters less than that called for by the specifications is not a good one, and should be discouraged, as it is the intention of the Navy Department that a uniform standard of test bar should be used, and this is not the case when bars of varying dimensions are submitted.

Physical Tests.—The main requirements of the physical test are elastic limit, tensile strength, elongation and contraction. In a specimen of good steel certain relationships of these terms may be expected; thus, the elastic limit should be at least half the tensile strength, and the elongation for two inches three-fifths to two-thirds the contraction. It will sometimes happen that a high elastic limit and high tensile strength are obtained, while the contraction and elongation figures are very low. In other words, the steel is strong but brittle. It will generally be found that the fracture in these cases shows more or less granulation; the silky appearance, so characteristic of tough steel that has been strained, being entirely absent. Again, the character of the fracture is generally that of a flat break, without any tendency to cupping. The cause of these peculiarities is probably the presence of too much phosphorus, carbon, sulphur, or manganese in the steel, or it may be improper annealing; an examination of the analyses in such cases will often show these elements in excess. It will be noted that the high-carbon steel produces sound castings, though they are frequently brittle.

In conducting the test care must be given to operating the testing machine. The speed must be slow until the point of elastic limit is reached, after which the speed may be increased, but it should be uniform at all times. The machine may be run by hand or by a motor; the latter, however, gives a more uniform speed.

Chemical Tests.—The limits set by the specifications for the elements phosphorus and sulphur must, in the main, be the guiding points for the inspector, and should be pretty closely adhered to; but it will occasionally happen that the service for which the casting is intended may permit of the specifications being slightly varied, and it is here that the inspector's judgment must be used. Experience shows that these elements can be controlled if the proper attention is paid to stock, especially in the acid process.

Annealing.—The purpose of annealing is to make the casting of uniform strength. The castings are brought to the required temperature of 660 degrees to 890 degrees C. and "soaked" twelve hours or longer, according to their character, and then gradually

cooled, the latter part of the process requiring as much care as the former, if not more.

All castings are required by the Navy Department to be annealed before testing. Low figures in the results of tests suggest at once that the annealing has not been sufficient; a second annealing, or even a third, may be necessary to bring the properties of a piece to the required standard. Although this requirement of annealing is made in Government work, it is not insisted on in the trade. Indeed, as a general rule, steel foundries do not anneal commercial castings, but nevertheless get good and satisfactory results. Still the practice of annealing is a good one and, in my judgment, should be retained, as it tends to make the casting more uniform in its qualities and undoubtedly better.

It has been claimed that steel can be produced by a secret process which does not require annealing, but this claim has not been borne out in the instances known. Steel making, as practiced today, is the result of long years of study and close observation, and when a claim is put forth that by means of a secret chemical mixture, introduced into the ladle just before pouring, sound castings and tough steel are assured, which require no annealing, it will be worth publishing the fact if such castings successfully pass the inspection tests.

Identification Marks.—An identification mark should be placed on all castings and coupons, when first submitted, with heat number, name, and number of piece, and a record made by the inspector of such submission; the record of each test should be preserved in the inspector's note book for reference in case another test from the same casting is required.

It generally happens, when a number of pieces are submitted for testing, that the heat numbers run along in a regular order. Suddenly the heat numbers of one or more pieces will appear out of this order. In such cases these pieces need looking after, and a search will generally show that they are second or third submissions of the pieces for testing. The foundry people may not be bound to point out this fact, but the inspector should know it, and he should depend upon himself to discover it.

Defects of Castings.—The general defects to be looked for in

steel castings are shrinkage cracks, particularly about angles; blow-holes, particularly on the upper portion of the casting; eccentric core-holes, due to shifting of cores; warping of casting, due to improper allowance for shrinkage, or improper cooling, and cracks due to not removing interior cores soon enough.

Hot and Cold Bending Tests.—In addition to the above tests, the Bureau of Steam Engineering, Navy Department, also requires the bending of the metal both hot and cold. The cold test is best made in the testing machine, using the forms furnished for the purpose; and the hot test is made at the blacksmith's forge, care being taken that the metal is not above a cherry-red heat and that the quenching water is clean.

IV.—STRUCTURAL STEEL.

In the inspection of mill and shop work the inspector should know what faults to look for; how, where, and when to find them; he should be thoroughly conversant with the methods of the shop or mill, arrange the inspection so as to follow the work in all stages of its progress, and keep himself posted as to what is being done in every department.

The sooner defective material or bad workmanship is detected the better, and the easier to remedy. The inspector should be familiar with the duties of the different men in the mill and shop, in order to take up points relative to rejections or improvements, and should see that necessary orders are given and carried out.

MILL WORK.

(1) *Study of Specifications.*—The specifications should be carefully studied for the character of the steel. The inspector should mark anything in them that is unusual or likely to cause extra work on the part of the mill to live up to, and he should consult on such points with the author of the specifications, so as to have a clear understanding of what is wanted before the work begins.

An abstract of the specifications should be made, showing physical and chemical requirements and tests required. This abstract should be copied in the note book for ready reference.

(2) *Order Sheets of Material.*—The draughting room should

supply a copy (in duplicate) of order sheets of material, containing estimated weights, and all information necessary to enable the mill to fill the specifications.

A great many draughting rooms make a practice of putting on each order sheet the part of the structure for which the material is intended. This is an excellent practice; it gives the draughting room very little extra work, and facilitates checking off, or referring to the materials.

The inspector, by knowing where material is to be placed in the structure, is in position to use some discretion, and will not reject material, such as filler plates, stiffeners, and like parts, on account of their deviating slightly from the requirements of the specifications. Work is often delayed and great inconvenience occasioned by the rejection of material possessing higher requirements than are necessary for the work it will be called upon to perform. On the other hand, he should mark on the order sheets the material on which the life of the structure depends, and insist on its filling the requirements in every respect.

(3) *Selection of Tests and Identification of Material.*—The inspector should ascertain from the mill what material for his work is rolled from each heat, and then select specimens for tests so as to represent the different sections rolled, for the working of the steel greatly affects the physical properties of the finished bar, thick metal giving different results from thin. It is his duty to satisfy himself that test specimens for the material are cut from sections of the same heat that they represent. All finished material should be stamped with the heat number of the ingot from which it is made, and when cut into shorter lengths these numbers should be reproduced on them, so that the heat from which each piece is made can be identified at any time.

(4) *Making Physical Tests.*—The test pieces must be properly prepared and of the size required by specifications.

(a) *Tensile Tests.*—In testing for ultimate strength and elastic limit, the machine, in order to insure accuracy, should be balanced before any testing is done.

The dimensions for the determination of elongation and contraction should be measured and the fracture observed. In case

a test piece should fail on account of a local defect, or on account of breaking in the grips of the testing machine, a re-test should be allowed.

(b) *Cold Bending Tests.*—The bending of test specimens can be performed in the way most convenient to the manufacturer, but they must be flattened down without cracking, to the amount required in the specifications.

(c) *Quench Bend Tests.*—In the case of quench tests, the material should be heated to the required degree and the water for quenching be of the specified temperature. The intention of this test is to show whether the steel, in case it should be heated to a red heat and suddenly cooled, would become so brittle as to render it unsafe. In some cases this test tends to water-anneal the steel, but as a rule it hardens it. If the test is conducted improperly, the steel would be either annealed or rendered worthless.

(d) *Hot Tests.*—The particular point to be watched is that the material is at the specified temperature while being bent or hammered.

(e) *Drift Tests.*—In making drift tests the hole should be punched at the specified distance from the edge of the piece to be tested, and a drift pin of proper taper used, noting to what size under various circumstances these holes may be enlarged without fracture of material.

(f) *Special Tests.*—Other tests sometimes required, such as opening and closing, flattening, breaking, torsional, impact, fracture tests, etc., must be made in strict accordance with the specifications.

(5) *Chemical Tests.*—The mill should supply a full chemical analysis of each heat, which may be checked at any time by making another analysis. But whenever an analysis is to be made for checking purposes, or simply to meet the requirements of the specifications, the drillings should be taken from the test bar, and the manufacturer be furnished a portion for a second or check analysis.

(6) *Surface Inspection.*—The extent of the inspection in the mill is mainly controlled by the specifications. Some specifica-

tions require the watching of steel from the time the raw material is put into the reducing furnace until it gets its final shape; and after it is rolled to its final shape, each bar must be turned and examined and the heat number identified.

In the case of structural material a more sensible method would seem to be to make surface inspection at the shop while the material is being worked into the structure, in which case the mill should be responsible for material rejected.

If each individual piece is not examined at the mill, each section should be inspected to see if it has been rolled true and to gauge, and to see that all fillets are well formed, that the web is smooth and free from buckles, and that there are no lumps or unevenness, due to defective rolls, that will interfere with the assembling. This inspection insures the section being good, and individual defective bars will be discovered and rejected during the shop inspection. In case bad bars are detected while material is inspected in lots, they should be thrown out at once, and if there are many defective bars all the material should be rejected, or each individual piece should be turned and inspected.

The following defects apply more particularly to shapes:

(a) *Defective Sections.*—Include shaded-down flanges, defective fillets, and the failure of the section to agree with the theoretical section.

(b) *Defective Edges.*—Include edges which are cracked from being burnt, rough and slivery, and wavy.

(c) *Surface Defects.*—The surfaces of sections, especially the webs of large beams and channels, are subject to the defects that occur on plates outlined under Section I, Steel Plates.

Other defects to be looked for are: (d) *Laps;* (e) *Bad Roughing Roll Marks* that have failed to weld in finishing passes, and (f) *Deep Scorings.*

(7) *Shop Work.*—(a) *Study of Blue Prints.*—Before the shop work commences, the inspector must be provided with a set of prints, approved by competent authority, or by the engineer in charge of the work. On the receipt of these, he should study the plans and obtain a clear idea of the structure in its entirety;

then study carefully all points and details in connection with the specifications and see that all notes on prints agree with them, for these notes are the instructions to the shop as to how the work shall be done.

He should make a memorandum, to submit to the engineer, of all points of disagreement between drawings and specifications.

In studying over the details, he should make notes on the prints of any points where difficulties in construction are likely to arise, and of such details as must be absolutely correct; he should devise methods of checking and insuring their accuracy. In case standard connections are not used (in beam and angle work), he should make a mark on the print to emphasize that fact.

Where sections are given in pounds per foot, he should put on the print the thickness, to facilitate checking up the sections during the progress of the work.

He should note on the prints the clearances allowed, so as to be sure the work will go together.

(b) *Preparing Material for Shop Work and Laying Out Work.*—All sections should be straight before any work is laid out to template. The template should be made of at least half-inch board, the parts being securely fastened together. When a member is being laid out the template must be in true alignment and firmly clamped to it. The center punch should fit the holes in templates snugly, and punch marks be well defined.

When the template is removed, all center marks should be marked with white lead, and the location marks put on the member.

(c) *Punching.*—The difference in size between the die and the punch should not exceed the following limit: one-sixteenth inch for punching metal up to one-half inch thick, and three thirty-seconds inch for thicker metal.

The punch and die should be well formed and smooth, and the punched holes should be free from jagged edges and excessive burring.

Where reaming is not called for, the material should be so punched that the burr end of the hole, wherever possible, will be on the outside after assembling. Where reaming is called for

the burr end of the hole is to be regarded as the punched diameter of the hole.

During the process of punching the inspector can detect any lack of uniformity, or undue hardness of the metal, by the way it punches, and can also determine whether the steel is being cracked or injured.

Any sections that have been bent by punching must be straightened, and any bad burrs or roughness of the section that will interfere with assembling must be chipped off. A piece of webbing should not be rejected because a punch hole has been wrongly placed, but the plugging of this punch hole must be done with hot metal; no condition allowing a cold plug to be driven.

(d) *Assembling.*—Before any work is assembled the abutting surfaces must be cleaned of lumps, rust, and dirt, and they must be well painted with specified paint; also all surfaces that will be inaccessible after assembling must be properly cleaned and painted. The inspector should bear in mind that this is the only protection these surfaces will ever receive, and that the proper painting of them is of more importance than the outside painting, which can be recleaned and painted when desired.

We should watch the work of assembling closely, and see that proper methods are employed to insure connections being properly located and adjusted. After the work is assembled and before any riveting is done he should see that all abutting surfaces are drawn tightly together and that the holes match, and that they are properly reamed, where necessary, so that there will be no shifting of the work by the use of drift pins.

In reaming holes the reamer should be held perpendicular to the face of the metal being reamed, and abutting members should be so tightly bolted together that no burr will be formed between them. In the case of machined surfaces where full bearings are required, the work should be drawn up tight, and the full value of the bearing obtained.

(e) *Riveting.*—The allowance for clearance in rivet holes is one-sixteenth inch, and this should not be exceeded. When working thick metal, the formed head end of the rivet should be

brought to a higher heat than the small end, so as to allow the body of the rivet to upset and completely fill the hole. Power-driven rivets should be used whenever possible, and enough should be tested on each piece to insure good work, but for the hand-driven rivets each one should be tested, as they are not driven under so great power and are usually so located that good riveting is hard to accomplish. Loose rivets should be cut out and replaced.

(f) *Facing.*—Facing should be done wherever called for on the drawing. If the surfaces are to be faced on a bevel, the bed of the facing machine is the best place to check the accuracy of the work. Where built-up sections are to be faced, the parts should be bolted in position as near the end as possible, and then faced.

(g) *Weighing.*—When the work is finished in the shop it should be weighed, and these weights checked on order sheets.

(h) *Cleaning and Painting.*—All steel and iron work must be well cleaned of scale, rust, dirt, and shop grease, and painted with the specified paint. The paint must be well rubbed in and all cracks and open places filled.

The inspector should witness as far as possible the mixing of the paint or employ a quick method of determining the character of that used, and make what analysis he considers necessary to determine its quality.

The subject of paints is a study in itself. It is taken up at greater length in succeeding pages.

(i) *Final Checking and Measurements.*—The inspector should make a final examination of the work, and assure himself that all the dimensions are correct and that the work will go together without trouble. In cases in which the work is very complicated it should be assembled at the shop, the necessary reaming and chipping done, and the different members match-marked.

Particular attention should be given to these points: The distance from the last hole to the end of the member, the chipping of countersunk rivets, the smoothness of bearing surfaces, and the proper finishing and smoothing up of slotted holes.

All through the shop work the note book on mill inspection

should be consulted to make sure that the steel tested is being used.

(8) *Shipments.*—No material should be shipped until after it is tested.

As material is shipped it should be checked off on the plans, and the material should be shipped in such a manner as not to delay the erection in the field. Often the failure to ship an important member will completely block the work of erection.

Copies of shipping bills, containing sections, weights, dimensions, and heat numbers, should all be examined and the actual weights of the finished material there shown compared with those estimated, so that the limit of tolerance is not exceeded. As these data are checked off on the order sheets the actual condition of the work as to material shipped and still due can be easily determined at any moment.

(9) *Inspector's Note Book.*—At the top of the page the inspector should put the name of the structure or job, and under this the order numbers that will be useful for reference. Then he should write an abstract of the specifications. He should leave the remainder of the page and the next page blank for any special remarks or modifications of the specifications. On one of the following pages he should make a classified list of material required; the different sections being placed in a column on the left side of the page, with the remainder left blank for inserting progress data, such as scheduled time for rolling, date of tests, heat numbers, etc. When all the material of a required section is rolled, he should run a pencil line through the item.

The advantage of a well-kept and simply-arranged note book is to add system to the work of inspecting and to enable the inspector, at any time, to ascertain the exact condition of the work in the mill.

V.—PAINTS AND PAINTING.

(a) *Cleaning of Steel.*—It is of prime importance that the steel be thoroughly cleaned before the paint is applied.

Steel when first rolled is covered with a mill scale which adheres to its surfaces with more or less tenacity. In the case of

heavy sections that are finished at a higher temperature this scale is loose and comes off in large flakes in the handling, but in spots it adheres and is not so easily removed. Light sections which are finished at a low temperature are covered with a closely-adhering scale, which is unbroken over the surface of the metal and forms, as it were, an enamel.

From the time structural steel is rolled until it has gone through the shop and been fabricated, several months as a rule have elapsed, and from handling and working most of the loose scale has dropped off; and in many cases steel has lain about the shop yards so long that it is covered with rust.

This rust is progressive under the original mill scale and, by growing beneath it, forces it off. In the case of a small section with mill scale closely adhering and forming a uniform coating, it will be found on exposure to weather that the rust will start at the edges and angles, and slowly extend itself beneath the scale, thus causing this original scale to fall off in flakes.

(b) *Cleaning of Shop Work before Painting.*—When the work is ready for painting, we have the following conditions: part of the steel is entirely covered with rust, the mill scale having been forced off by its growth; part is covered with rust and mill scale adhering more or less tightly; and part is free from rust but has on it the original mill scale. All is more or less covered with dirt and grease.

The best that can be done in the shops at present, to prepare this steel for the paint, is to give it a good cleaning with hammer, steel scrapers, and wire brushes. The sand blast would be better, but very few shops are equipped for this.

The shops will slight as much as possible any cleaning of material before the paint is applied, so the inspector must watch this point very closely and insist upon all steel being freed of rust and scale as thoroughly as possible.

(c) *Relation of Paint to Scale and Rust.*—When steel entirely covered with rust is rubbed with wire brushes and scrapers, it has a dirty and muddy appearance, for it is impossible by this means to get it entirely clean. A surface of this kind, however, takes the paint well and looks well after painting. The paint

combines with the rust, and it is an open question whether or not the rust will be progressive beneath the paint.

When the steel is covered with both rust and scale, any scale adhering after scraping is very undesirable, for under it rust has formed more or less, and the oil of the paint is absorbed by the small particles of this rust, and the combination of oil and rust causes an expansion which loosens the scale and causes it to drop off. I have seen cases in which freshly painted work looked very well, but in a few days the paint and scale came off in large flakes, leaving bare spots on the metal.

When the steel is entirely covered with mill scale, that which is loose is easily removed; the rest adheres tightly. If this adhesive scale is covered over properly with paint, in all probability it will continue to stick for a long time. But the trouble here is to avoid having weak places in the paint around the edges of the scale. Any break in the paint around this edge will be a sure opening for rust to attack.

Another place very susceptible to rusting is about rivet heads. In driving the rivet the mill scale in the immediate vicinity is cracked, and on account of the rivet head being in the way it is very hard to clean off this cracked scale, which adheres more or less tightly. Then the painters are very apt to leave little open spaces in the paint about the rivet heads. In reports on the paint of structures after several years' exposure it will be noted in all instances that the most rust is found about the rivet heads. The same is true in the case of boilers; yet the experts do not attribute it to the cracking of the scale by riveting, but explain it as the effect of electrolysis.

Another case in which the cracking of the surface scale is seen to aid rusting is when a thin plate, entirely covered with a closely adhering scale, is bent cold and then entirely painted. This plate, if exposed to the weather, will begin in a very short time to rust at the bent portion. An example of this is seen in coal buckets and other articles made of thin steel plates.

I have dwelt at considerable length on this subject of scale and rust, for to my mind it is the principal cause of most of the failures of good paints.

(d) *Application of Paint.*—All paints should be thoroughly stirred and mixed just before use, and all surfaces on which the paint is to be applied should be clean and dry. The inspector should see that all abutting surfaces, and surfaces not accessible after assembling, are painted with the specified paint. These surfaces, as a rule, are not extensive, and the tendency of the shop men is to use the paint that is nearest at hand.

After the shop work is completed and the material thoroughly cleaned, the method of application of the paint is to be watched. All edges and ends of sections should be well painted; all open spaces and joints should be thoroughly protected; and special care should be taken in painting well about rivet heads. The paint should be put on uniformly and no part of the material left uncovered.

(e) *Kinds of Paints Used.*—The paints generally used for structural steel are graphite, iron oxide, and red lead; the base for all should be pure linseed oil.

(f) *Graphite Paints.*—The graphite paints are generally the output of some standard manufacturer, and are shipped to the shops already mixed and ready for application; therefore, about all the inspector can do is to see from the tags on the barrels that a paint of a reputable manufacturer is used; or, if a special brand is specified, that this particular paint is being applied to his work.

The vehicle of the best graphite paints is pure linseed oil, and the pigment is a graphite, either pure or combined with silicates, carbonates, or combinations of other elements, depending upon the experience or fancy of the manufacturer. Each manufacturer claims some secret which makes his paint greatly superior to that of his competitors.

There is a vast difference in graphite paints; but, in the present state of the paint industry, it will take some time for any one of the leading brands fully to demonstrate its superiority.

The Detroit Superior graphite, Dixon, No-brack, Protectus, and a half-dozen other makes are all good graphite paints.

These paints originated with the lead-pencil manufacturers and furnished means for them to dispose of waste products. The

pure graphite is light, flaky, and greasy; hard to grind to fine powder, and is not a scientific paint. These difficulties are met by a great many paint people by combining the graphite with other elements, such as calcium, silicates, etc.

(g) *Iron Oxide Paints.*—These paints range from a perfectly worthless article to a very good paint. They are generally mixed by the manufacturers and shipped in barrels to the shops.

The cheap grades of this paint are not much more than a mixture of red mud and mineral oil, or cheap fish oil.

The best iron paint is the oxide (Fe_2O_3) produced by burning Ferrous Sulphate, ground in pure linseed oil.

Mineral or fish oil can be detected in paint by the odor, or by the way it burns when ignited. This test can be applied by dipping a piece of paper into the paint, withdrawing and setting fire to it. Linseed oil sputters a great deal when burning and gives off but little smoke, while the other oils burn readily and give off a black smoke.

The determination of the character of the iron pigment is a question for the chemist. The inspector can, however, depend upon it that if iron oxide paint is specified, the shop will use about the cheapest grade.

(h) *Red-Lead Paint.*—There are many substitutes on the market for red-lead paint, and much of the low-grade and adulterated product is sold. The architect, or engineer, before writing his specifications, should acquaint himself with the purity and fineness of the different brands manufactured, and specify by name. As a specification calling for "red lead" may mean, so far as the inspector is concerned (as he has no ready means at hand of determining), a lead oxide that contains anywhere from thirty to fifty per cent. litharge (PbO), he has to accept a red lead that will simply pass the blowpipe test. This may be "pure lead" that has not been properly fired, or, in other words, has not received the highest form of oxidation, which is very essential, in order that it may have the proper affinity for linseed oil which it will "take up" in its "set" on the steel, after application. The amount of linseed oil the red lead will take up in setting is from twenty-three to twenty-five per cent. of its weight.

Any oil in excess of this will cause the paint to streak on the surface of the steel. The simple formula of from thirty to thirty-three pounds of red lead to one gallon (7½ pounds) of pure linseed oil, is based on a scientific fact and must be followed in order to obtain the best possible results. Pure red lead cannot be mixed with linseed oil and shipped any great distance without "setting" in the package, so beware of already mixed red lead or red-lead paste, as any substance, liquid or solid, that will hold red lead in suspension in the package will prevent its taking the proper "set" on the steel, after being applied.

The paint must be mixed fresh every morning, and well stirred, and must be constantly stirred while being applied, to prevent the red lead from settling to the bottom of the buckets.

The painter is very likely to omit this stirring, in which case most of the red lead settles to the bottom of the bucket and he paints with a discolored linseed oil.

When red lead has been mixed for several days before use, it takes its "set," and, when applied, runs in streaks on the painted surface and loses much of its value as a protective coat.

Any drier is injurious to the paint. Pure red lead with raw linseed oil, if properly applied and in the proper proportions, dries in about twenty-four hours, and gives a pretty and shiny surface. The use of benzin or turpentine in the paint cuts the oil and prevents the proper "set," causing it to dry quicker, but gives the paint a dead appearance and is injurious to it.

Red-lead paint can be easily tested by the use of the blow-pipe. If it is pure, a lead button, free from any other substances, will be the result. If mixed with impurities, these impurities cannot be gotten rid of by the blow-pipe flame and are left as a residue on the charcoal. There are very neat little blow-pipe outfits for the testing of paints, which can be carried in the pocket.

A quick method to test red-lead paint is to stir it well, dip a match in it, withdraw and light it. As the match burns it will reduce the paint on it to pure lead. By holding a piece of paper under the burning match small particles of pure lead will be seen to drop from it on the paper. If there are impurities in the paint, the lead cannot be so readily reduced.

Where pure red lead and pure linseed oil of old and well-established brands are specified the inspector can see it mixed and can then be sure of the character of the paint used.

(i) *Linseed Oil is used either Boiled or Raw.*—Boiled linseed oil is more or less of a trade name and is generally looked upon with doubt unless it is the product of some reputable manufacturer.

There are so many ways of diluting boiled linseed oil, and the impurities are so hard to detect, that it is better not to specify it.

RAW LINSEED OIL.

Oil freshly pressed, designated as new oil, has a greenish color, and is unfit for use in paint. Old oil is generally yellow, except when bleached, in which case it has the appearance of water. The old, or yellow oil, is the kind that should be used in mixing paints. Too much stress cannot be laid on the importance of the use of pure linseed oil in paints, for the oil is the vehicle for the pigments and binds them together. The best pigments mixed with a poor oil will make a worthless paint, while almost any pigment mixed with pure linseed oil will make a good paint.

The common adulterants of linseed oil are petroleum, corn and cottonseed oils, rosin, and fish oils. Cottonseed and corn oil never dry properly, and they cause the paint to be "tacky" and the surface to blister.

Petroleum and fish oils dry slowly by evaporation, causing the painted surface to be soft for a long time, and when it does finally dry the surface of the paint is porous and lusterless and very likely to crack.

Rosin oils dry slowly, leaving a brittle paint of a short life. Any one of the above adulterants, if mixed with linseed oil, causes the paint to partake of the character of the adulterant.

Pure linseed oil dries, not by evaporation but by the absorption of oxygen, which causes it to contract and become tough and tenacious.

The following are methods of testing linseed oil.* Take ten

*U. S. Medical Dispensatory.

parts of the suspected oil, mix in a small cylindrical glass tube with three parts of crude nitric acid, agitate the mixture well and allow it to rest. If it is pure, it becomes, during agitation, first sea green and then a dirty greenish yellow; after settling, the nitric acid has a bright yellow color.

Linseed oil is frequently adulterated with fish oil, also with cod-liver oil, both of which adulterants are cheaper than the pure oil of the flax. These adulterations may be determined by the following tests:

(a) Dip the finger into the oil and spread over a smooth surface, glass preferably, and if the oil contained either of the mentioned adulterants, it will not dry.

(b) Pure linseed oil has a pleasant, bland taste, and the presence of fish oil may be readily detected by its nauseous effect.

Linseed oil may also be tested for purity by boiling, and if, while boiling, the raw oil gives off any scum, it is a sure indication of the presence of impurities. Boiled oil cannot be tested in this way, and, in fact, is hard to test any way.

The importance of selecting the right paint and the proper application of it on the thoroughly cleaned surface cannot be too strongly put. The result of slighting the painting of steel, before it is enclosed in a building, is very dangerous.

CONCLUSION.

The inspector should always have the work well in hand and be observant; if there is any fault to be found with the way the work is being done, he should speak of it to the right person, so as to have the remedy applied in a proper manner.

He should be courteous, but firm; always mindful of his duty; he should not expect perfect work, but do everything in his power to obtain the best results, and make the work a credit to all concerned; and he should remember that it is better to be respected for conscientious work than to cater for friendships at the expense of his own reputation.

He should work with a view of increasing his own knowledge and gaining in expertness. He should make notes of what he observes and of all experience gained on each piece of work.

I wish to express thanks to Mr. E. McL. Long, and P. A. Engineer Andrew McAllister, U. S. Navy (retired), assistant inspectors in the office of the Inspector of Engineering and Ordnance Material for the U. S. Navy, of the New York District, for valuable contributions and assistance in the preparation of these notes.

I also wish to express my appreciation of suggestions made by numerous professional friends.

SCREW PROPULSION FOR NAVAL AND MARITIME PURPOSES.

By George Quick, Fleet Engineer (Retired), H. B. M. Navy.

SYNOPSIS.

1. Introduction.
2. Advantages of large helicoidal area in steaming against head winds, &c.
3. Necessity for further towing experiments to ascertain the resistance of deep-draught ships at high speeds in deep water.
4. Reference to influence of depth of water on resistance of ships at high speed.
5. Proposed formula for calculating maximum speed for minimum resistance for ships of various lengths and different coefficients of fineness.
6. Proposed formula for calculating maximum speed for any given maximum of resistance. Reference to Cunarders of 750 feet length.
7. Speculative calculation of H. M. S. *Drake* at various speeds from 9 knots to 26 knots.
8. Proportion of helicoidal surface to maximum I.H.P. to secure good efficiency.
9. Skin resistance, edge and thickness resistance of propeller blades, and reference to trials of propellers with two to six blades.
10. Friction of engines and load.
11. Power expended in producing recoil of water at various speeds.
12. Pitch ratio.
13. Power required to overcome rotational resistance of propellers. Empirical formula for calculating same.
14. Advantages of wide ends to propeller blades for high speeds.
15. Advantages of curved receding blades for high speeds.

16. Advantages of triple propellers, the endurance of propelling power for naval and maritime purposes, and of interchangeability of all parts of the machinery of ships of the same class.

INTRODUCTORY.

In the preparation and presentation of the following brochure, the primary object sought was to urge naval administrators as well as naval architects and marine engineers to the importance and necessity of giving increased consideration to the study of the action and theory of the screw propeller. The tendency of the times, both as regards naval and maritime construction, calls for increased speeds of rotation of the propeller, as well as for increased power transmitted per unit of propeller-blade surface.

During the past thirty years both tendency and practice have been to use propellers with a form of blade quite narrow at the tips. There are now indications that progressive development along naval and maritime lines calls for a return to the earlier design of propeller which possessed fan-shaped blades. In order to further increase efficiency, the blades should be given a curved receding form so as to reduce the resistance due to rotative action.

As the rotative velocity of the screw has been increased, the form of the surface which experiences rotative resistance should be modified to suit the higher speed conditions. As the bow lines of ships are modified to suit required speeds, it is quite certain that extended experiment would demonstrate that for definite speeds that modifications in the form of the blade would likewise secure increased efficiency of the propeller.

The power expended in driving a tail-race directly rearward is an absolute definite expenditure, but it will be excessive in amount if the propellers have too large area of blade surface for the diameter used. On the other hand, when the blade is too small in area, either for the power applied or for the resistance of the ship, there is a stripping of the thread of the water. Such action will cause loss of propulsive efficiency due to an insufficient quantity of water being forced astern.

Some rotation of the water is inherent to the action of the screw. This produces a loss of efficiency which cannot altogether be avoided, but improvement could be effected by giving the blade surface a curved and receding form so as to drive the water outwards toward the ends of the blades.

For the past forty-five years the writer has been deeply interested in the problem of the screw propeller. Previous to entering the Royal Navy of Great Britain in 1860, the subject had appealed very strongly to him. The first naval vessel to which he was assigned was H. M. Gunboat *Bullfinch*, stationed at Portsmouth Dockyard, which vessel was used for some years in testing various propellers proposed by rival inventors. Among the screws fitted to this ship and which were reported upon were those of Griffiths, Hirsch and Révy.

At the same time the British Admiralty were conducting experiments with Griffiths propeller fitted on H. M. S. *Cygnet*. H. M. S. *Shannon*, of 3,800 tons displacement and 2,000 I.H.P., was likewise assigned to the duty of testing ordinary screws having two, three, four and six blades. During this period experiments were also made with the French Mangin screw at the Portsmouth Dockyard. The writer not only carefully studied the results of those trials, but conferred with officers who helped conduct the tests. The interest in the problem of the screw propeller which was aroused in the early years of his naval career, has continued to the present day, and this paper therefore represents the study, observation and experience of many years.

It should be stated that the observation and consequent conclusions arrived at do not apply to small vessels or to torpedo boats and destroyers, but to ships of heavy tonnage and deep draught. Both warships and merchant vessels which make extended voyages must be prepared to encounter head winds for about one-third of the time they are under way, and that propeller will be the most efficient which will give good results even when steaming against head winds and heavy seas.

Even as early as 1863–64 the writer's attention was specially directed to the problem of the propeller by noting the comparative performances of H. M. S. *Racehorse* and a French ship of

similar size and power,—two vessels which were frequently called upon to make the round trip between Shanghai and Yokohama. In fine weather, H. M. S. *Racehorse* made the better passage, but when encountering rough weather the French ship showed a decided superiority. The only reason that could be found for the general superior performance of the French ship was that she had a screw propeller of much larger area than that of the similar English vessel.

The ability of a vessel to maintain good speed against head winds is a factor of great value to sea-going people, and this element of value is seldom discovered during a trial trip or by experiments with models.

Discussion on the screw-propeller problem has been comparatively fruitless during the past forty years. In fact, so little definite knowledge has been secured that hope of further improvement appears to have been abandoned by many naval architects and marine engineers. The succession of failures from propellers which were specially designed either to increase the speed of the ship or to secure economy of fuel at moderate speeds has caused many to believe that progressive improvement in the design will be exceedingly slow.

Extended experiments with different forms of propellers should be made upon large ships, and it would then undoubtedly be found that the results secured would be quite different from that obtained with models. In tank experiments the investigator cannot possibly secure the conditions that exist at sea, and therefore the results secured will not apply to the vessels when put to actual work.

The striking success which has attended the endeavor to increase the speed of the Whitehead torpedo has been due to the fact that the torpedo itself has been experimented with and not miniature models. It is simply because the large vessels have not been used for experimental purposes that a corresponding improvement has not been effected in steamship propulsion.

Since the admirable experiments of the late Mr. W. Froude on H. M. S. *Greyhound* we have had no towing experiments on a large scale to ascertain the actual resistance of ships at high

speeds and in deep water except those conducted by Mr. Yarrow on a torpedo-boat running at 15 knots speed.

Tank experiments, whether conducted in Europe or America, cannot and have not produced data or results that merit comparison with information secured from experiments made on large-sized ships in deep water. Experiments with models may indicate the direction in which investigation should be afterwards pursued with the ships. They cannot be relied upon for the formation of hard and fast rules governing the design of either ships or propellers. It is actual experiment, experience and observation with ships that give absolute information. Possibly the best confirmatory evidence of this fact is the knowledge that the most valuable contribution which has been made of late years to the subject of the resistance of ships was the discovery of the influence of depth of water on the speed of vessels, and this fact could only have been ascertained from actual experience with the ships. The cases of H. M. S. *Blenheim* and the U. S. S. *New York* are worthy of notice. So also is that described by Capt. Rasmussen, Engineer-in-Chief of the Danish Navy, published in the Transactions of the Institution of Naval Architects for 1899. That paper shows clearly how much the depth of water affects the apparent resistance of the vessel at different propelling powers.

Another cause of lack of definite information as regards propulsion is due to the fact that it has become customary for the resistance of ships at the highest speeds to be calculated from the gross I.H.P. It has thus been contended that the resistance of the ship increases with enormous rapidity when the limiting speed is exceeded, so that at 12 per cent. above that limiting speed the I.H.P. required varies as the seventh power of that speed. This point is admirably brought out in a valuable paper by Mr. Tennyson d' Eyncourt, on "The Limits of Economical Speed of Ships," published in the Transactions I. N. A. for 1901, and reprinted in the August, 1901, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, and by Mr. James Hamilton in his paper on "Horse Power Absorbed by Skin

Friction and Wave Making &c.," published in the Transactions of the I. N. A., 1898.

The writer is compelled to differ somewhat from the conclusions arrived at by those gentlemen. The resistance of the ship and the power required to overcome that resistance do undoubtedly rise rapidly after the limiting speed has been reached, but may not a larger part of the excess of I.H.P. required to obtain those high speeds be due to the increased waste of power by the screw propellers at the higher rotation? This is a point to which we will have occasion to return later on. But that the British and foreign shipbuilding authorities are opposed to such a view of this matter is indicated by Mr. Hamilton, who says in his paper: "There is good reason for believing that the percentages of the total indicated horsepower representing propeller waste and load friction are less in high-powered vessels than in low-powered." That is quite true when the I.H.P. per square foot of helicoidal surface of the screw does not exceed certain limits, but it is impossible to find any evidence that the propeller losses do not increase more rapidly than the resistance of the ship when certain limits of I.H.P. to helicoidal surface are exceeded. Yet it is generally assumed that the propeller losses do not amount at the highest speeds to more than 31 per cent. of the I.H.P. or 51 per cent. of the net propelling power.

This is a matter not for logic or dispute, but a question that can only be settled by direct experiment on a full-sized ship with engines and propellers working at the highest speed. The recent trials of H. M. S. *Drake*, and the other large cruisers, *King Alfred*, *Good Hope* and *Leviathan*, so far as their results have been published, have afforded an opportunity for investigating—

(1) The resistance of ships at various speeds.
(2) The expenditure of engine power at the various speeds.
(3) The irreducible non-productive expenditure of power.
(4) The reducible non-productive expenditure of power.

These investigations are far from being as complete as one would like to have them, but they are submitted in their rough condition in the hope that a discussion may lead to far higher speeds being attained than have hitherto been regarded as pos-

sible. For these formulæ and calculations no claim is made that they possess the accuracy of pure mathematics, or that they are applicable for all classes of vessels, but they are submitted in the belief that they will merit close examination by those who are interested in increasing the size, speed and economy of our ships of war and commerce.

MAXIMUM SPEED OF MINIMUM RESISTANCE.

Taking the results of the towing trials of H. M. S. *Greyhound* as a basis of calculation we have—

L_1 = length = 172 feet.

C_1 = coefficient of fineness = .575.

S_1 = speed before resistance increased more than as $S_1^2 = 8$ knots.

For the new cruisers of the *Drake* class I assume—

L_2 = 500 feet.

C_2 = .525.

S_2 = speed to be ascertained before the resistance increases more rapidly than S_2^2.

Then

$$S_2 = \frac{L_2^{0.83} \times C_1 \times S_1}{L_1^{0.83} \times C_2}.$$

$$S_2 = \frac{500^{0.83} \times .575 \times 8 \text{ knots}}{172^{0.83} \times .525}.$$

$$S_2 = 21.0 \text{ knots}.$$

That indicates that the speed of the *Drake* would be 21 knots before the resistance would begin to increase faster than as S_2^2. The analysis of the *Drake's* trials shows a similar result.

Applying this empirical formula to the proposed new subsidized British cruisers of 750 feet length, and supposing them to have the same coefficient of fineness as the *Drake*, we find that their speed would be 29.7 knots before their resistance would increase faster than as S^2. But if their coefficient of fineness were .675, then the highest speed of minimum resistance would be only 23.2 knots.

MAXIMUM SPEED AT FIXED MAXIMUM OF RESISTANCE.

From the *Greyhound* experiments it appears that up to 8 knots the resistance varied as S^2, and that at ten knots $R_1 \propto S^{2.825}$ it is required to find the speed of the *Drake* before the resistance R_2 will $\propto S_2^{2.825}$.

Using the same values as before, except that

$$S_1 = 10 \text{ knots},$$

we have

$$S_2 = \frac{500^{0.83} \times .575 \times 10 \text{ knots}}{172^{0.83} \times .525}.$$

$$S_2 = 27 \text{ knots}.$$

Thus at 27 knots the resistance of the *Drake* will vary as $S^{2.825}$, and the net horsepower for propulsion only should vary as $S^{3.825}$. But by reference to the Table I, line 4, it is seen that the I.H.P. between 20.03 knots and 22.16 knots varied as $S^{4.16}$. With the original blades, of 152 square feet total for two screws of helicoidal area, it appears that between 22 knots and 23 knots in the *Drake*, the gross I.H.P. varied as $S^{6.736}$; in the *Leviathan* as $S^{5.636}$; the *Good Hope* as $S^{7.716}$, and *King Alfred** as $S^{3.8}$, and in the *Kent* as $S^{7.3}$. This discrepancy disproves the theory of the increase of G.I.H.P. being entirely dependent on hull resistance at the higher speeds, and that the hull resistance increases suddenly as S^7 or S^{10}.

Applying this to the proposed cruisers of 750 feet length, we find the speed at which the resistance will vary as $S_3^{2.825}$, thus:

$$S_3 = \frac{750^{0.83} \times .575 \times 10 \text{ knots}}{172^{0.83} \times .525}.$$

$$= 37 \text{ knots if coefficient of fineness were } .525;$$
$$\text{or } 34 \text{ knots if coefficient} = .575;$$
$$\text{or } 28.9 \text{ knots if coefficient} = .675.$$

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* The *King Alfred* had screws of increasing pitch, which gave lower coefficients of performance than the *Drake's* screws, of uniform pitch and of 210 square feet of helicoidal area. See Table, page 666.

These are rough approximations of the speeds we may expect to get in very large ships when our methods of propulsion have been improved to such an extent as they certainly might be by judicious experiments. We shall certainly find that it is not the resistance of the ship that increases rapidly to the seventh or tenth power of the speed, but that it is the screw propellers which fall off in efficiency at the higher speeds with terrible rapidity, and that thus the larger part of the G.I.H.P. is spent uselessly. The writer is aware that the results given above do not accord with the results deduced from model experiments, but until towing experiments on large ships in deep water have proved that the resistance of long, fine ships increases suddenly as the fourth to the tenth power of the speed, he cannot but regard the above result as being nearer to the truth than the results obtained from models.

SPECULATIVE CALCULATION OF RESISTANCE OF H. M. S. *DRAKE*.

The following is a rough attempt to calculate the resistance of the *Drake* from the results of the trials of the *Greyhound* and other ships.

R_4 = resistance of *Greyhound* at 4 knots, or S_4.
R_6 = resistance of *Greyhound* at 6 knots, or S_6.
R_8 = resistance of *Greyhound* at 8 knots, or S_8.
R_{10} = resistance of *Greyhound* at 10 knots, or S_{10}.
D_1 = displacement of *Greyhound* = 1,160 tons.
C_1 = coefficient of body = .575.
L_1 = length = 172 feet.
K = a constant number = 9.5.

Then

$$R_4 = \frac{S_4^2 \times D_1^{.666} \times C_1}{L_1} \cdot \frac{}{K}$$

$$R_4 = 16 \times .369 \div 9.5.$$

$$= 16 \times .03884.$$

$$= .62 \text{ tons.} \quad \text{By experiment } R_4 = .60 \text{ tons.}$$

$$R_6 = 36 \times .03884.$$
$$= 1.398 \text{ tons.} \quad \text{By experiment } R_6 = 1.4 \text{ tons.}$$

$$R_8 = 64 \times .03884.$$
$$= 2.4857 \text{ tons.} \quad \text{By experiment } R_8 = 2.5 \text{ tons.}$$

At 10 knots the resistance of the *Greyhound*

$$R_{10} = S^{2.083} \times .03884 = 121.1 \times .03884.$$
$$= 4.7 \text{ tons.} \quad \text{By experiment } R_{10} = 4.7 \text{ tons.}$$

Resistance of *Drake* at various speeds:

$$R_{Speed} = \frac{S^x \times \frac{D_2^{.666} \times C_2}{L_2}}{\frac{D_1^{.222}}{D_2^{.222}} \times K}.$$

The value of x varies from 1.90 at the lower speeds to 2.005 at 22 knots, 2.015 at 24 knots, to 2.04 at 26 knots (see line 14), and to 2.083 at 27 knots.

The resistances at the various speeds are shown in the Table No. 1, line 16. They range from 8.68 tons at 9.5 knots speed, to 73.2 tons at 24 knots and 92.4 tons at 26 knots. They are calculated from $S_k^x \times 0.12 = R$ $(0.12 = K_1)$.

These resistances have been calculated with the view of ascertaining the highest limit of speed that we may hope to gain from practically perfect propellers. But whatever the actual tow-rope resistance would be found to be, if the *Drake* could be towed at all speeds up to 27 knots, it is very certain that there is a large amount of power still lost by the latest propellers tried on the *Drake*, and that there is ample room for improvement if proper means are taken to obtain it, and are not dominated too much by the ideas of the past and by the results of model experiments.

PROPORTION OF HELICOIDAL PROPELLING AREA TO I.H.P.

Nothing is more striking than the difference in practice respecting the proportion of I.H.P. to the square foot of helicoidal

area, projected area and disc area in these days from that which existed in former years. In H. M. ships, between 1855 and 1870, the proportion ranged from 6 to 12 horsepower per square foot of helicoidal area and projected area.

In the *Iris* and *Mercury*, in 1878, there were 14 I.H.P. per square foot of disc area with the screws of 18 feet diameter, and 37 I.H.P. per square foot with the screws of 16 feet diameter, at maximum speeds and highest powers.

In the third-class cruisers of the *P*, or *Prometheus* type there are about 37 I.H.P. per square foot of disc area, but there are 148 I.H.P. per square foot of helicoidal area.

In the second-class cruisers of the *Hyacinth* type there were 150 I.H.P. per square foot of helicoidal area. And in the case of the latest and largest cruisers, *King Alfred*, *Good Hope*, *Leviathan*, and with the original screws of the *Drake*, there was about 200 I.H.P. per square foot of helicoidal area. The later screws of the *Drake* are 210 square feet in helicoidal area and have 150 I.H.P. per square foot of helicoidal area.

In the United States Cruiser *Columbia*, of 22.8 knots speed and 18,000 I.H.P., there are 112 I.H.P. per square foot of helicoidal area.

The proportion of helicoidal area to I.H.P. to secure the best results and the highest possible speed is worthy of most serious attention—far more than it has yet received in the provision of screw propellers for war ships and fast, high-powered mercantile steamers. In the case of H. M. S. *Drake* the greatest efficiency was attained between 15 and 20 knots when there were only 30 to 70 I.H.P. per square foot of helicoidal surface. As the proportion rose to 150 and 200 I.H.P. per square foot of helicoidal surface so the efficiency fell off. The decline in efficiency at the higher speeds is ascribed entirely to the enormous increase in the wave-making resistance of the ship. The writer, however, maintains that a very large part of the decline in efficiency is due to the enormous waste of power due to insufficient helicoidal area of blades generally, *and to insufficient width at the end of the blades, and to the radial form of the blade surface being*

unsuitable to the high speed of the propellers, as was pointed out in a letter to the Admiralty in June last. The proportion of I.H.P. to area of helicoidal surface must, of course, vary in ships of various block coefficients and proportions of beam and length. The longer the ship and finer the lines of the ship, the larger may be the proportion of I.H.P. per square foot of helicoidal surface, but in no case is it desirable that there should be more than 130 I.H.P. per square foot of helicoidal surface in sea-going ships having a coefficient of fineness of about .5, and for ships having a coefficient of fineness over .6, the proportion of I.H.P. to helicoidal should be approximately 70 to 1.

FRICTIONAL SKIN RESISTANCE, EDGE AND THICKNESS RESISTANCES OF PROPELLER BLADES.

This matter received much attention from Mr. Isherwood of the United States Navy, from the late Mr. Blechynden and many others. The author agrees with those who consider that the increase of area, *per se*, does not lead to a large and useless expenditure of power. In H. M. S. *Bullfinch*, in 1861–62, a three-bladed propeller was substituted for a two-bladed propeller of practically the same diameter, pitch and length, but the three-bladed screw had about 50 per cent. more surface. The results were as follows:

.....	Revs.	I.H.P.	Speed.	$\frac{S^3 \times \text{I.M.S.}}{\text{I.H.P.}}$	Slip.
					per cent.
Two-bladed screw.....	195	254	8.545	324	23
Three-bladed screw.....	185.5	249	8.541	329.5	17

There was no loss here from increased friction surface and edge resistance. Experience at sea shows that with head winds and rough water the three-bladed propeller with its 50 per cent. greater area would have shown a more decided advantage over the two-bladed screw.

The case of H. M. S. *Shannon* in 1862, with screws 18 feet diameter, was still more striking, the results being as follows:

Description.	Pitch.	Revs.	I.H.P.	Speed.	$\frac{S^3 \times D^{\frac{5}{3}}}{\text{I.H.P.}}$	$\frac{S^3 \times \text{I.M.S.}}{\text{I.H.P.}}$	Slip.
							per ct.
Two blades.....	23.2	58.6	2,057	11.28	164	522	17.8
Mangin	25.1	53.2	2,033	11.33	168	534	14.0
Two blades.....	24.9	57.4	2,096	11.08	153	485	21.0
Four blades.....	23.8	53.2	2,023	11.55	179	569	7.0
Two blades.....	24.9	57.7	2,084	11.01	151	478	22.0
Six blades.....	23.7	49.7	1,956	11.21	169	538	3.2
Three blades....	23.7	56.1	2,055	11.49	175	555	12.0

Thus the four-bladed screw gave the highest maximum speed and the most economical results. The proportion of I.H.P. to helicoidal surface was as 20 to 1. The helical velocity at the end of the blades was only about 33 knots against about 70 knots in the case of high-powered modern propellers, but my investigations show that in the U. S. Cruiser *Columbia* and other modern ships that the power required to overcome the resistance due to the skin friction, edges and thickness of the blades does not amount to more than four per cent of the I.H.P. at any speed.

When the pitch and diameter remain the same in two similar screws but having different areas of helicoidal surface, the revolution, N, will vary inversely as the .123 power of the areas thus:

$$N_2 = \frac{A_1^{.123} \times N_1}{A_2^{.123}}.$$

The marked improvement in the speed of the *Drake*, from 23.03 knots to 24.11 knots by the increase of the helicoidal area from 152 square feet to 210 square feet, is a further proof that the truth established by the trials of the *Bullfinch* and *Shannon* still holds good today. Four per cent. has therefore been allowed for the propeller friction, etc., in these calculations.

FRICTION OF ENGINES AND LOAD.—INTERNAL FRICTION.

This is variously estimated by different authorities. It undoubtedly varies considerably in different engines and under different conditions of the same engines and ship, as, for instance, when the ship is weak in her structure and there is excessive vibration.

Under fair circumstances 10 per cent. of the I.H.P. is a just allowance for the combined engine and load friction for modern engines having all the latest improvements in design and workmanship and the great increase in modern initial and mean steam pressures. In the United States triple-screw ship *Columbia* the thrust horsepower was calculated to be nearly 82 per cent. of the I.H.P. In the calculations of the results of the *Drake* it was assumed that 86 per cent. of the I.H.P. is delivered to the screw for propelling the ship and disturbing the water, after allowing 10 per cent. for engine friction and 4 per cent. for unavoidable propeller resistance. These are denoted in Table I as P_e, for the power to overcome engine friction, and P_f for the power to overcome friction and resistance of the propellers, and P_n indicates the net shaft power = 86 per cent. of the I.H.P. at all speeds.

POWER EXPENDED IN PRODUCING RECOIL OF WATER.

Not much use is here made of the popular term "slip" in connection with screw propulsion, for reasons too numerous to mention. It is well known that the best results are not always obtained by the propellers which give the smallest amount or the largest amount of "slip." That propeller is best which gives the largest value at all speeds to C_S and C_D when

$$C_S = \frac{S^3 \times \text{I.M.S.}}{\text{I.H.P.}},$$

and

$$C_D = \frac{S^3 \times D^{0.666}}{\text{I.H.P.}}.$$

If we can increase the value of C_S and C_D at all speeds, *and especially at the highest possible speeds*, it is not a matter of economic importance whether the resulting apparent "slip" be 5 10, 20 or 30 per cent.

Now there is a certain proportion of power which is indicated by P_w (in the Table I), which must be expended on the water whatever the velocity of the feed or of wake, or whatever the eddy motion may be. We cannot control the motion of the

water around the screw unless we can alter the form of the stern lines, which is practically impossible. But if the propellers be designed on the very simple and practical principle of making the helicoidal area to bear a certain liberal proportion to the maximum I.H.P., say one square foot to 70 I.H.P., we have the power in our hands to reduce that surface at a very small expense of money and time in the event of our finding that the required I.H.P. and speed of ship be not obtained in consequence of the pitch and area of the blades being too great for the engines. It is possible that a reduction of pitch may give the increased number of revolutions and increased speed required, but if that reduction of pitch were not found sufficient to get the required power and speed, then a reduction of the area of the blades, either by reduction in diameter or in width, could be effected at very small cost in time and money. The writer believes that a very great improvement would be made in the average passages of all our high-powered steamships if the helicoidal area of all the propellers were increased by 30 or 40 per cent., by the substitution of four-bladed screws for three-bladed ones. But great judgment would be required to design the screws of four blades to get the best possible results. A mere rule-of-thumb application of the principle would be attended with results perhaps not worth the cost of the alteration.

The expenditure of power in producing the water recoil P_w, is calculated from the simple formula:

W = weight in tons of water acted on per minute.
= (disc area — boss area) × pitch × revolutions per minute ÷ 35 cubic feet.
v = velocity of water acted on in feet per minute.
D = displacement of ship in tons.
V_h = speed of ship in feet per minute.

Then $$v = \frac{D \times V_h}{W}.$$

The proportion of power P_w expended in producing this unavoidable recoil of the water to the power P_p expended in over-

coming the tow-rope resistance of the ship is given by the formula:

$$w = \frac{W \times v^2}{D \times V_h^2}.$$

$$P_w = \frac{W \times v^2 \times P_p}{D \times V_h^2} = P \times w = \text{H.P. spent on recoil.}$$

Referring to Table I, line 20, it will be noted that the proportion of power P_w to P_p continuously decreases as the speed of the ship increases until at the highest speed it is only about 34 per cent. of P_p. There are good reasons to suspect that by a slight alteration of the blade surface, the power expended on P_w may be very slightly diminished at all speeds. But that is a matter to be practically tested by experiment in a vessel of high speed and in deep water, and any discussion thereon at present would be futile.

PITCH RATIO.

This subject has been so thoroughly explored by Mr. R. E. Froude, by Mr. S. W. Barnaby, and more recently by Mr. McDermott, of New York, that the writer would only add the opinion that by certain alterations of the form of the blades we may make screw propellers of larger pitch ratio than at present, and probably showing a larger amount of apparent slip, and yet giving a higher maximum speed to the ship, with higher values to the coefficients C_S and C_D. That also is a matter for experiment and not for present discussion.

POWER REQUIRED TO OVERCOME ROTATIONAL RESISTANCE.

To ascertain approximately the proportion of power expended in overcoming rotational resistance it is desirable to consider extreme cases.

Consider the most simple case, that of a Smith's, or common screw, 10 feet diameter, 2 feet length; diameter of boss, 2 feet; fitted to an imaginary ship 420 feet long, 30 feet beam, 15 feet draught; coefficient of fineness, 500; imaginary engines, 30,000 I.H.P.; immersion of center of screw, 7.5 feet. The ship is free in deep, smooth water, no currents. Pitch of screw infinitely

great. What will be the I.H.P. when the screw is driven 100, 150 and 200 revolutions per minute? What will be the maximum number of revolutions per minute when the I.H.P. is 30,000?

Practically all the power will be expended in overcoming the rotational resistance of the screw, for there will be no speed ahead. In this case, P_r, which represents the power expended in overcoming the rotational resistance in the Table I, line 25, will be = 1.00; and P_p, which represents the useful propelling power, will be = 0.

Consider another screw, of the same dimensions, but of 31.4 feet pitch, to be substituted for the first screw, and find the I.H.P. when this propeller is driven at an equal number of revolutions to the former. Note, also, the mean revolutions per minute when the I.H.P. is 30,000. The ship will have considerable speed ahead, and if the tow-rope resistance of that vessel at that speed be known, the value of P_r and P_p may be calculated. Repeat these operations with screws of pitch ratio 2, 1.5, 1.25, and 1.0 and from the results the proportion of P_r to P_p may be approximately calculated for any shape of blade.

Experiments with a 420-foot ship such as those above referred to have, of course, not been made, but, investigating this matter as far as means will allow, it appears that the proportion of power expended in overcoming the resistance to rotation at normal pitch ratios, say, under 1.5, increases at high speeds much more rapidly than the power expended in producing propulsion of the ship and in producing water recoil. (See Table I, lines 17 and 25.) This point is clearly shown by analysis of the trials of H. M. S. *Kent*, from the data published in the "London Times" of 20th of February, 1903.

To ascertain the best form of blade, the writer designed in the first place a model with pliable blades of infinitely great pitch. These blades were therefore planes at right angles to the plane of revolution, and consequently produced no propelling effect, all the power applied being expended in overcoming their rotational resistance.

These pliable blades could be bent by hand to any form desired, and by curving the blades backwards from the direction of

rotation, the rotational resistance at a given number of revolutions was diminished up to a certain limit approximately as the cosine of the angle of recession. But it is obvious that the blades may be so curved that while the rotational resistance is thereby diminished the propelling efficiency may be reduced also.

The problem then was to ascertain the form of curve for screws of ordinary pitch ratio which, with the minimum of rotational resistance, should give the greatest propelling thrust for the power applied. That form of blade gave a divergent sternward current, and drew its feed supply from forward of the screw.

The ordinary, or Griffiths propeller blades are normal, or at right angles to the axis of revolution; that is, the screw surface is practically that of a square-threaded screw, and consequently experiences a large amount of rotational resistance, especially at high speed of revolution and fine pitch ratio, and particularly in cases where the pitch ratio is too great for the form of the ship. Many hook-shaped blades have been designed during the last fifty years, such as the Nystrom and Hirsch blades, to gain an increase of propulsive efficiency; but the increase of the rotational resistance of those hook-shaped blades and disturbance of the currents flowing to and from their propelling surfaces have caused the efficiency of those propellers at high speed to be less than that of the ordinary screw.

It is evident that, whether a true-screw surface of metal be curved, V-shaped or square, the travel in a metal nut will be the same if the pitch be the same; but in the case of water or air we have a yielding fluid to deal with, which fluid admits of slip. But we have not to consider so much the percentage of slip, but the more important matter of the maximum speed of ship that can be obtained, and the value of the coefficients at all speeds.

These coefficients being

$$\frac{S^3 \times \text{I.M.S.}}{\text{I.H.P.}}, \text{ and } \frac{S^3 \times D^4}{\text{I.H.P.}},$$

the power P, line 25, required to rotate the propeller with the proposed form of blade becomes $P \times$ cosine of angle of recession, while the propulsive efficiency is not reduced, the surface

near the periphery being increased in area and the curvature at that part being arranged to get the best results for propulsion.

When we increase the pitch of a screw of ordinary design beyond a certain narrow limit, we do not increase the propulsive forward thrust to the same extent that we increase the resistance to rotation.

Take any very small surface of the driving face of the screw where the pitch angle is less than 45 degrees, and compare the circular or rotational velocity with which it moves with the forward velocity of the same small area in the direction of propulsion.

The following shows approximately the circular velocity of a small patch of the propelling surface of the *Drake's* screws (of 210 square feet helicoidal area) at different diameters, when the speed of the ship and forward speed of the screw surface were 17.93 knots and 24.11 knots. The screws are 19 feet in diameter.

Diameter.	Forward speed of ship and of propelling surface.	Circular or rotational speed of propelling surface.	Forward speed of ship and of propelling surface.	Circular or rotational speed of propelling surface.
feet.	*knots.*	*knots.*	*knots.*	*knots.*
18	17.93	48.8	24.11	68.2
16	17.93	43.4	24.11	60.6
14	17.93	38.0	24.11	53.2
12	17.93	32.6	24.11	45.8
10	17.93	27.1	24.11	38.4

The helical velocities of the propelling surfaces are slightly in excess of the circular velocities, and are used in calculating the frictional edge and thickness resistances of the blades.

Mr. Tennyson d'Eyncourt, in his paper on "The Limit of Economical Speed of Ships," states that he found from actual trial results that "at about 12 per cent. above the limiting speed it (the I.H.P.) is varying as the seventh power of the speed, whilst the wave horsepower varies as v^7 at the limiting speed, and as v^{10}, or sometimes as a higher power of v, at 12 per cent. above the limiting speed." Those figures are, of course, calculated from the gross I.H.P. of the ships tried, and actually represent the facts *as they appear;* but, as they are not calculated on tow-rope experiments on full-sized ships of deep draught at the speeds

referred to, they cannot be accepted as being perfectly correct indications of the growth of the hull resistance alone at those speeds.

According to Joessel's formula, given in Taylor's "Resistance of Ships," page 27, the resistance of a plane of area, A, moving perpendicularly to itself in sea water becomes, for speed in knots,

$$R_c = 4.62\ A.V^2.$$

From this the wave resistance of one foot square at 20 knots would be 1,848 pounds; at 40 knots, 7,392 pounds; at 60 knots, 16,632 pounds, and at 70 knots, 22,514 pounds.

Now the screw surface does not move perpendicularly to itself, but at angles, varying from 45 degrees near the boss to about 21 degrees at the periphery. From the time of Bossut, in 1777, and Beaufoy, in 1798, to the present day, no experiments appear to have been made on the resistance of planes at all angles from 0 degrees to 90 degrees, and at speeds from 10 to 70 knots; and therefore for determining the resistance of planes at high speeds the experiments hitherto made are almost useless. But we may, without great error, allow that the resistance of a plane at a given angle will vary as some power y of the velocity V, and that as the velocity increases beyond certain limits the value of the index y will probably increase also.

Now, it has been calculated from the I.H.P. that at about 12 per cent. more than their limiting speed the power required to overcome the total resistance of ships increases as S^7, and the power required to overcome the wave-making resistance increases as S^{10}, that is, say, when a ship 400 feet long is driven at 22.5 knots and over, or a 750-foot ship at 30.6 knots.

The bow and stern lines of very few steamers have so great an angle as 17 degrees, which is the smallest pitch angle that any ordinary propeller has at its periphery. It is therefore not unreasonable to suppose that the wave-making resistance of the surface of the propellers (which surface makes angles varying, in the case of the *Drake*, from about 21 degrees at the periphery to about 45 degrees near the boss,) at circular speeds ranging from 38 knots, near the boss, to 68 knots, near the periphery,

when the ship is going at the highest speed, will vary quite as much or more than the resistance of the hull at about one-half or one-third of the circular speed; that is, from 18 to 24 knots.

From many small experiments, and also from analyses of the trials of ships and propellers at the highest speeds, the writer has arranged an empirical formula for calculating the power which is expended at all speeds for overcoming the rotational resistance of the screws. The formula as proposed includes full consideration for the varying angles of the propelling surface and its varying velocities, although it is somewhat simpler than the formula given by Mr. Calvert in the 28th volume of the Transactions I. N. A. Without going into detail regarding the means by which this formula has been constructed,

Let G = the rotational speed in knots of the radius of resistance of the propelling surface.

y = variable index of the speed G. (It is to be noted that this index y for the rotational power required is of very nearly the same value as the index, $x + 1$, for the resistance of the hull. See lines 14 and 23.)

$K_2 = 0.06$, a constant derived from experiments.

Then the horsepower required to overcome the rotational resistance of the screws at various speeds will be

$$P_r = G^y \times K_2 = G^y \times 0.06.$$

It will be noted by referring to lines 21 and 25 that the proportion of power expended on overcoming the rotational resistance of the screws rises rapidly with the increase in the number of revolutions of the screws. *This is a matter of paramount importance*, and one to which too little attention has been paid hitherto. The writer has given many years to the solution of this problem, that is, to the study of the reduction of the rotational resistance of propellers and the utilization of the power so saved to the direct propulsion of the ship. But experiment alone on a ship at the highest powers can prove the advantages of this theory, for such advantages cannot be shown at low powers and low speeds.

1. Area of helicoidal surface of blades, in square feet	152	152	152
2. Speed of ship in knots	15.43	22.08	23.05
3. Gross I.H.P.	6,937	23,103	30,557
4. P varies as S^m between adjacent speeds. Value of m	3.357	6.736	
5. P varies as S^n between 15.4 kts. and higher speeds. Value of n		3.719	
6. I.H.P. per square foot of helicoidal surface $= P \div A = Q$	45	152	203
7. Revolutions per minute. N	72.3	105.9	116
8. Velocity of screws, feet per minute. Pitch $\times$ revolutions $= V_s$	1,771	2,595	2,842
9 Velocity of hull in feet per minute. V_h	1,563	2,237	2,335
10. Slip, per cent	11.8	13.9	18
11. Estimated friction of engines and load=10 p. c. of I.H.P. $= P_e$	694	2,310	3,055
12. H.P. calculated to be spent in overcoming skin-friction, edge and thickness resistance of screws $= P \times .04 = P_f$	277	924	1,222
13. Net horsepower supplied to screws for propelling the hull and disturbing the water $= P - P_e - P_f = P_a$	5,966	19,869	25,780
14. Tow-rope resistance of hull varies as S_h^x. Various values of x	1 95	2.007	2.01
15. Value of S_h^x for various speeds	207	498	548
16. Tow-rope resistance of hull in tons $= S_h^x \times 0.12 = R$ (in tons)	24.92	59.78	65.78
17. Horsepower expended in propelling hull $= R \times V_h \div 14.73 = P_r$	2,644	9,010	10,427
18. Weight of water in tons, acted on by screw blades per min. = disc area − boss area $\times$ pitch $\times$ revs. $\div$ 35 cu. ft. or W	25,666	37,594	41,180
19. Velocity imparted to the water, W, in feet per min. $v =$ Displacement $\times V_h \div W = v$	852	833	794
20. Energy of water recoil + energy of hull $= W \times v^2 \div D \times v_h^2 = w$. D is the ship's displacement in tons	.544	.374	.34
21. Horsepower expended in producing the recoil of the water $= P \times w$	1,438	3,370	3,540
22. Circular or rotational speed, in knots, of radius of resistance of blade surface $= G$	31	46	50
23. Power spent in overcoming rotational resistance of propellers $= P_r$, and varies as G^j. Value of j	2.9	3	3.01
24. Value of G^j for the various rotational speeds of the screws' faces. G^j	21,131	97,336	130,000
25. Horsepower expended in overcoming rotational resistance $= P_r = G^j \times .06$. P_r	1,268	5,840	7,800
26. Horsepower accounted for in propelling hull, in water recoil and in rotational resistance $= P + P_e + P_r = P_a$	5,350	18,220	21,772
27. Difference between P_a and $P_s = P_l =$ H.P. lost or gained in "churning" due to any defect of design of propellers. P_l	616	1,649	4,008
28. Ordinary coefficient of performance $S^2 \div$ I.H.P. $=$ coefficient $= C_a$	.53	.463	.428
29. Efficiency calculated by W. Froude's *Greyhound* formula resistance $\div$ ind. thrust. (Resistance here $= R$.) Coef. $= C_b$	.432	.456	.415
30. Absolute efficiency by W. Froude's formula E.H.P. $\div$ I.H.P. $= C_d$	.381	.390	.341
31. Proportion of power spent on rotation to net power, P_a, of screw, when $P_a = 1.0$. $P_r \div P_a = C_d$	.212	.294	.302
32. Proportion of power spent on rotation to power spent on propelling hull, when $P = 1.0$. $P_r \div P = C_e$	.476	.648	.748

a. Quick's screw with four curved and receding blades, 280 sq. feet.....*b*. Area of Quick's blades, 33 p. c. greater than *Drake's* present blades, but max. No. of revolutions maintained by reduction of resistance to revolution by curvature and recession of surface.....*c*. $G^j \times .03052$.

210	210	210	210	210	210	210	*a.*
9.52	13.06	15.41	17.93	20.03	22.16	24.11	26
1,885	4,014	6,520	9,872	14,801	22,534	31,409	31,409
3.251	2.93	2.74	3.66	4.16	3.94		...
...	...	...	2.74	3.07	3.41	3.51	...
8	19	31	47	70	107	150	112
47	64	75.1	87.6	98.5	111.5	122.4	122.4 *b.*
1,076	1,468	1,727	2,015	2,266	2,565	2,815	2,815
964	1,317	1,560	1,816	2,030	2,244	2,442	2,634
10.4	10.3	10.4	9.8	10.4	11.24	13.21	6.4
168	401	652	987	1,480	2,253	3,140	3,140
67	160	261	395	592	901	1,256	1,256
1,450	3,453	5,607	8,490	12,729	19,380	27,013	27,013
1.9	1.9	1.95	1.95	2	2.007	2.015	2.04
72.33	132	207	278	401	502	610	770
8.68	15.84	24.84	33.36	48.12	60.24	73.2	92.4
567	1,416	2,630	4,113	6,632	9,177	12,137	16,523
15,650	21,312	25,010	29,170	32,800	37,130	40,760	40,760
860	865	873	872	866	846	840	904
.893	.657	.559	.48	.426	.377	.345	.343
506	930	1,470	1,974	2,825	3,460	4,187	5,667
20	28	33	38	43	48	53	53
2.9	2.9	2.9	2.9	2.9	3	3.015	3.015
5,929	15,550	25,330	38,140	54,560	110,592	158,000	158,000
358	933	1,519	2,288	3,273	6,653	9,480 *c.*	4,823 *d.*
1,431	3,279	5,619	8,375	12,732	19,290	25,804	26,930
19	174	*e* 12	115	*e* 3	70	1,209	Nil. *e.*
.512	.553	.561	.584	.540	.479	.446	.559
.377	.394	.446	.462	.502	.465	.44	.555 *f.*
.336	.353	.403	.416	.448	.407	.386	.526 *g.*
.247	.27	.271	.27	.257	.343	.351	.178
.631	.659	.577	.556	.493	.725	.781	.292

d. Power required to overcome rotational resistance with Quick's blades = $P_2 \times$ cos of angle of recession = 4,823 H.P.....*e* Apparent gain or excess, probably due to errors of data or of calculation.....*f*. The highest values for *Greyhound* and *Mutine* were .517 and .457.....*g*. The highest values for *Greyhound* and *Mutine* were .422 and .363.

ORDINARY RADIAL SURFACE SCREWS OF THREE OR FOUR BLADES OF MAXIMUM EFFICIENCY AT HIGHEST SPEEDS.

If we have a screw or screws of good pitch ratio and one whose helicoidal area is properly proportional to the horse-power, then, to get the highest speed and maximum of efficiency, we should have

$$P_p + P_w + P_r = P_n,$$

and there will be no loss of power from "churning" or loss due either to insufficient or to excess of blade area. When such unnecessary expenditure of power appears, as it does in some of the trials of the *Drake* and other modern ships, it is represented as P_l in the Table I, line 27.

This is the useless expenditure of power which can be easily avoided by judiciously modifying the form of the blades, and which it is proposed to reduce in all existing ships *by increasing the width of the blades at the end* and by substituting four-bladed screws of 33 per cent. greater helicoidal area for the three-bladed screws with narrow blade ends, as at present used.

But after the total elimination of the loss P_l the question arises,—Cannot the expenditure of power P_w in producing the recoil of the water, and the expenditure of the power P_r in producing only the rotation of the propellers, be diminished, and the power so saved be utilized in increasing the value of P_p so as to get a much higher speed of hull?

The writer has made many small model experiments, but he considers all such experiments as giving merely indications of the lines on which to pursue investigations and experiments. No definite laws can be deduced from these model experiments on the reduction of the resistance to rotation of curved or receding blade surfaces, such as are here proposed. So far as experiments and speculative investigations can go, they show that *by reducing the width of the blades very considerably near the boss and by curving and receding the blade surface as described* the resistance to rotation may be so diminished that from thirty to fifty per cent. less power will have to be expended on rotation only, and that the power so saved from reduced resistance to rotation may be expended—

(1) In giving a higher number of revolutions to a curved bladed screw of pitch, diameter and area equal to the ordinary screw with radial blades, and a higher speed of hull; or,

(2) That the screw with the curved receding blades, if of the same diameter and pitch, may have the width of the end of the blades very much increased without the revolutions of the engines and the maximum power of the engines being increased or reduced thereby, *and that such increased area at the end of the blades and reduction of width near the boss would lead to a considerable reduction in the number of revolutions required per minute to obtain any required speed;* or

(3) For very long ships of fine lines these curved receding blades with wide ends may have the same diameter and total area as ordinary blades, but a higher pitch ratio, in order to obtain high speed of hull with a moderate number of revolutions per minute. If in this case it were found that insufficient revolutions, I.H.P. and speed were secured, the question whether the pitch should be decreased, or the blades reduced in diameter, could be ascertained from calculations made on the results obtained by that particular ship and propeller.

The total changes in screw-propeller design that the writer would propose to effect are somewhat wider than those here indicated. It should be remembered that a change of propellers can be effected at less cost than any changes in connection with boilers and engines, and that such changes in the screws cannot give rise to any trouble after they have been adopted. Every saving of power and increase of speed by the screw propeller which may be effected is of the most far-reaching value for naval and mercantile purposes, and cannot introduce any countervailing disadvantages, such as sometimes attend the introduction of new forms of engines or types of boilers.

The writer is aware that in proposing to increase the width of the end of the blades he is reverting towards the earlier form of propeller, and abandoning the form introduced by the late Mr. Griffiths. But he would point out that the conditions now are different from what they were forty years ago; also *that the locality of the propelling surface is of as great importance as the*

proportion of the area of that surface to the maximum I.H.P. One square foot of surface at or near the periphery is equivalent to at least *eight* square feet at or near the semi-radius of the blade.

It is impossible for propeller blades with narrow ends and great width near the boss, and whose superficial area is 100 square feet, to give as great speed and efficiency by making more revolutions, as would be secured from another propeller of the same pitch and diameter, possessing less total helicoidal area, *but having greater width at the ends of the blades and less width near the hub.*

It has required many years of scientific research and hundreds of trials to bring the ordinary radial-surfaced screw to its present condition of efficiency, and it is therefore hoped that the system here proposed may have some preliminary trials in order to develop its highest possibilities.

Vague and indefinite changes should not be made, but the experiments should be progressive so that the development would be along conservative lines. Extended experiment and observation would particularly show where details of improvement could be effected.

MULTIPLE PROPELLERS.

Multiple propellers were employed nearly fifty years ago, and the turbine system which is in some respect a development of the idea is bringing them again into notice. For nearly thirty years the writer has advocated triple propellers, *with three isolated engine rooms*, in order to obtain during battle as much security as possible for some portion of the propelling power. There are to the sea-going engineer a dozen good reasons for the adoption of triple propellers in large high-speed ships for every single objection urged against it. It cannot be supposed that certain foreign powers, such as Russia, France and Germany, have been hasty and injudicious in adopting triple propellers for their large warships. Germany alone has nearly forty large warships built and building with triple propellers, and it is to be hoped that the British Admiralty may soon give triple propellers a trial in some very large ships. Admiral Melville of the

U. S. Navy has so admirably stated the case for triple screws that not much of importance can be added to what he has said.

The introduction of triple screws in vessels where there is an installation of over 20,000 horsepower is just as logical a development of machinery design as the introduction of twin screws was a generation ago.

In the case of twin screws it was found that the tactical and strategic advantages surpassed the economical and structural benefits that were looked for. And so it has been with the triple screws. The naval strategists of Germany, Russia and France now find that the tactical advantages alone are sufficient to demand installation of three screws, since ships with triple screws have a maneuvering power that cannot be secured with ships fitted with only two screws. Then, again, the art of steering is acquired with triple screws in less time than either with single or double screws.

With reference to triple screws for war vessels, however, and especially for battleships, may it not be of advantage that the center propeller should have only two blades, for the following reasons:

(*a*) That when the center engines are not working that propeller may be placed in the vertical position, and that in that position it will neither retard the vessel so much, nor require so much power to drag it, as when a three or four-bladed screw was used. Furthermore, it would not interfere so much with the steering of the vessel as a three or four-bladed screw would do either when the engines were going ahead or backing. Particularly in the case of towing or when approaching or leaving a dock would the advantages be apparent for the middle screw to have two blades.

(*b*) That the two-bladed center propeller shaft being kept coupled, that engines could be used at immediate signal. Furthermore, for battleships under 30,000 I.H.P, it would be well to allocate two-fifths of the power to each wing screw and one-fifth of the power to the center screw, and this small portion of power to the center screw would give a speed of nearly 12 knots to ships of the *Duncan* and *Russell* class, or 15 knots to the large cruisers of the *Drake* and *Kent* classes.

(*c*) With the propeller shaft of the two-bladed center screw always connected to the crank shaft, it would not be necessary, as suggested by some, to install small engines for rotating the center propeller when that screw was not in use. Neither would it be necessary to fit any special gear to the other main engines to drive the center screw.

The arrangement of two four-bladed side screws and one two-bladed center screw would give ten blades for propulsive effect instead of the six blades now generally employed in most of the twin-screw ships.

For very large naval ships and subsidized mercantile cruisers, such as the proposed Cunarders, the triple screws would all have equal engine power, and the center screws would not need to remain idle, as these ships would generally have to steam at speeds exceeding 18 knots, and therefore the center screw as well as the two wing screws should have four blades each. By this proposed arrangement the propelling power in these ships would be exerted by twelve separate blades, instead of six blades only, as at present in twin-screw ships having three-bladed screws.

In war service we must expect that the propellers may be and will be much more frequently damaged than in ordinary cruising service. Accidents to propeller blades of warships and mail steamers occasionally occur even under ordinary circumstances, and when one blade of a three-bladed screw is broken off or seriously damaged the remaining two blades will, if that screw be used for propulsion, throw a very heavy and unequal stress on the shafting and cause excessive vibration of the ship. If one blade of a four-bladed screw be broken, only one-fourth of the propelling surface will be lost (instead of one-third), and less stress will be thrown on the shaft. Less vibration will also be produced if the impaired screw be worked, so that a higher speed may thus be obtained than with the two remaining blades of a three-bladed screw. This is a point worthy of the consideration of executive officers who may have to command fleets in future battles, and one that also should not be ignored by ship owners.

In consequence of the four blades of a propeller, *if arranged*

in the ordinary manner, being nearer together, that is, 90 degrees apart (instead of 120 degrees, as in three-bladed screws), it is possible that at the highest speeds the action of the four blades may cause such a disturbance of the water, or cavitation, as to diminish the efficiency of propulsion at the highest speed in some cases.

This subject of the interference of multi-bladed screws and their excessive disturbance of the water causing cavitation was made a subject of special duty by the writer in 1861 and 1862, in connection with the propeller tried by Mr. Révy, which had practically twelve blades surrounded by a casing forming part of the screw and, of course, revolving with it. As a result it is believed that means can be devised to prevent the interference of the blades with each other.

A naval sea-going engineer can think only of the necessities of the day of battle and the week afterwards. He places supreme value on the endurance of his boilers, engines and propellers for the rude, hard work and stress of long-continued steaming in war, both before and after the day of conflict. He realizes that everything should be subordinated to preparing for the exigencies of actual war.

Forty years ago, after the conclusion of the war in China, the writer was appointed to a gunboat which had been four years in commission, and under steam for 296 days during the previous year, without any repairs having been made in the meantime in a dockyard. The machinery and boilers were in fair working condition, although both the working and spare propellers were broken. This made it necessary to get a spare propeller from another vessel, and one was secured which exactly fitted, thanks to the principle of interchangeability which had been introduced for those engines by Mr. John Penn during the Crimean war. That old gunboat remained in commission for three years more, and afterwards took part in the bombardment of Shimonisaki, before she finally went to Hong Kong dockyard for refitting.

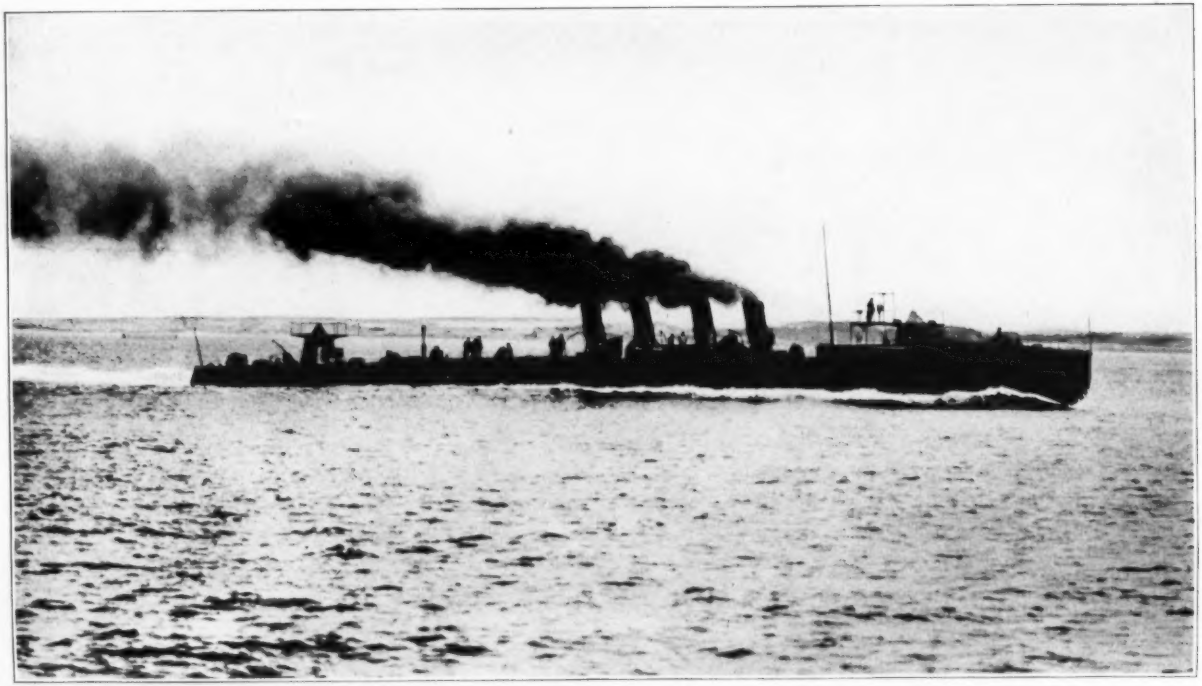
Fifty years of experiments have failed to produce any substantial improvements in screw propellers, and nothing less than a series of experiments organized on scientific and practical prin-

ciples can prove whether the designs herein proposed would break through this hitherto impenetrable barrier to progress and show the possibility of improvement.

Surely the facts herein presented would warrant the expenditure of a sum sufficient to carry on an extended series of tests. The question of the propeller is one worthy of the consideration of the Admiralty of every leading power, and some of the cruising ships of each nation could be employed in no better way than by making comparative tests with screws of various forms. In addition to the efficiency and economy that would be secured, such experiments would prove of immense benefit to the executive officers in teaching them how to handle and maneuver more efficiently the great floating fighting machines which have been entrusted to their care, and for whose safety and efficiency the nation at large holds them responsible.

TABLE OF RELATIVE COEFFICIENTS $\frac{S^3}{I.H.P.}$.

.....	Speed.	Coef.	Speed.	Coef.	Speed.	Coef.
Good Hope.....	15.91	.506	22.1	.480	23.05	.394
Leviathan.....	15.23	.546	21.96	.462	23.25	.397
King Alfred.....	15.16	.543	21.59	.446	23.46	.419
Drake (original).....	15.43	.529	22.08	.463	23.05	.428
Drake (new).....	15.41	.561	22.16	.478	22.16	.446



U. S. S. "LAWRENCE," TORPEDO-BOAT DESTROYER, AT 27.27 KNOTS.

CONTRACT TRIAL OF THE UNITED STATES TORPEDO-BOAT DESTROYERS *LAWRENCE* AND *MACDONOUGH.*

By Lieutenant Walter Ball, U. S. N., Member.

The *Lawrence* and *Macdonough* are twin-screw sea-going torpedo-boat destroyers, contracted for by the Fore River Engine Company, at Weymouth, Mass. (now the Fore River Ship and Engine Company, Quincy, Mass.), from their own plans and specifications, the price being $281,000.00, each, and the time allowed for completion 16 months, to be computed from December 3, 1898. The price above stated does not include the ordnance, ordnance outfit and a few articles supplied by the Government.

The vessels were to be standardized up to the highest attainable speed, which could not be less than 28 knots, and, after being satisfactorily standardized, run for at least one hour at an average speed of not less than 26 knots. In case the vessels did not make the required speeds, the question was to be dealt with upon its merits.

The air pressure in the fire rooms was to be sufficient to maintain the maximum rate of combustion. The following weights were to be carried in addition to that of the complete hull and machinery, spare parts and water necessary for the trial, viz: torpedo gear and torpedoes, rapid-fire guns and ammunition, 15 tons; coal, 34 tons; crew, boats, anchors, electric plant, equipment, etc., 12 tons; total, 61 tons. When thus loaded the draught amidships was 6 feet 6 inches and 8 feet to clear propellers.

The weight of the machinery was limited to 194 tons, including main engines, boilers and appurtenances, auxiliaries, steam heating appliances, water in the boilers, condenser, pipes and feed tanks, and such stores and spare parts as are carried on board when fully equipped, but did not include stores supplied by the Government, steam steering gear, steam windlass, dynamos and engines, and torpedo air-compressor machinery. The penalty

U. S. S. "LAWRENCE," TORPEDO-BOAT DESTROYER, AT 27.27 KNOTS.

for overweight was $200.00 per ton for over 194 tons, and if this exceeded 5 per cent., then $1,000.00 further was to be deducted from the contract price of the vessel.

HULLS.

The hulls are constructed of both mild and nickel-steel of domestic manufacture, with frames spaced 20 inches apart, where the intermediate frames are placed in engine space, that is, from No. 80 to No. 105, they are to be arranged to suit the engines. The outer keel is 12-pound plate, the vertical keel is 15-pound plate, the garboard strake is 9-pound plate, tapering off to 8-pound plate fore and aft; the main plating is 8½-pound plate, tapering off fore and aft to 7 and 6-pound plate; the sheer strake is 12-pound nickel-steel plate amidships for two thirds of the length, tapering off to 9 and 8-pound plates of carbon-steel fore and aft.

The ships are divided into twenty-three watertight compartments.

Hold.—In the hold abaft the engine compartments and forward of the boiler compartments are magazines, trimming tanks, store rooms and fresh-water tanks.

Forward Platform.—On the forward platform are the windlass engine, engineer's room, ward room, armory, pantry, bath room, staterooms, captain's room and petty officers' quarters.

After Platform.—On the after platform are the galley and crew's quarters.

Coal Bunkers.—There are four coal bunkers, having a total capacity of 116 tons at 43 cubic feet per ton. They are filled through scuttles in the main deck.

The fire rooms may be cleared of ashes by ash ejectors, situated one in each fire room, or by buckets through the hatches by means of fall and tackle.

Drainage System.—For draining the compartments there is a 7-inch main drain of copper, which leads from the forward boiler room to the after engine room. This pipe is fitted in order to enable large quantities of water to be removed by means of connection with the main circulating pumps in the engine rooms.

There is also an auxiliary drain leading from forward of frame 11 to aft of frame 124. This pipe is 3¼ inches O. D. from the forward boiler room to the after engine room. Forward and aft of these spaces it is 2½ inches O. D., all of copper. These suctions are placed as low as possible in the boats. The ends of the suctions are covered with galvanized wire strainers.

In the engine rooms the 7-inch main drain is fitted with strainers.

Each large compartment is fitted with a 2-inch Hancock steam ejector having a steam pipe ½-inch diameter, there being seven ejectors in the boat.

Waste water from bath and wash bowls is led by pipes directly overboard in most cases.

The boats carried are as follows:

Two 20-foot cutters. Two 18-foot folding boats.

One 20-foot whaleboat.

HULL DATA.—TRIAL LOAD.

Length between perpendiculars, feet and inches	246– 3
on L.W.L., feet and inches	240–10
over all, feet and inches	246– 3
Beam, extreme, feet and inches	23– 3
Ratio of length to beam	10.94 to 1
Depth in hold from top of main deck beams to keel, feet and inches	19– 9
Draught, forward, seagoing trim, feet and inches	5– 1⅛
aft, seagoing trim, feet and inches	7– 9⅝
mean, seagoing trim, feet and inches	6– 5½
Displacement, seagoing trim (load draught), tons	400
per inch at L.W.L., tons	8.7
Area of immersed midship section, square feet	106
Center of gravity of L.W.L. plane, aft of midship section (frame No. 74), for even keel, feet	12
Center of buoyancy above bottom of keel, feet	3.875
aft of midship section, feet	6.6
gravity above bottom of keel, feet	8.34
Transverse metacenter above center of buoyancy, feet	7.5
Longitudinal metacenter, above center of buoyancy, feet	8
Coefficient of fineness on extreme dimensions	.42
of midship section	.81
L.W.L.	.68
Cylindrical coefficient	.54
Number of frames	146
watertight compartments	23

ARMOR.

The forward conning towers are $\frac{3}{4}$-inch nickel-steel of 90,000 pounds tensile strength, and the after conning towers are $\frac{1}{4}$-inch carbon-steel.

ARMAMENT.

The battery for each boat consists of

Two 3-inch semi-automatic rapid-fire guns.

Five 6-pounder semi-automatic rapid-fire guns.

Two tubes for 18-inch Whitehead torpedoes, on center line of boat, aft.

MAIN ENGINES.

There are for each vessel, two vertical, inverted, direct-acting, four-cylinder, triple-expansion engines placed in separate athwartship, watertight compartments. The starboard engine is in the forward compartment. The sequence of the cylinders is H.P., I.P., 1st L.P., 2d L.P.; the H.P. being forward. The sequence of cranks for ahead motion is H.P., 2d LP., I.P. and 1st L.P. The cranks are placed at angles of 90 degrees. The framing consists of forged-steel columns braced by diagonal and horizontal fore-and-aft and athwartship braces. The columns are secured to cast-steel bridge pieces, which form supports for the crank-shaft bearings. The bridge pieces are in turn secured at their ends to fore-and-aft angles 8 inches by 4 inches by $\frac{3}{4}$ inch, which are secured to the engine seatings forming part of the hull. The cylinders are not jacketed and have no working linings, consisting of a cylindrical casting with a cylinder attached for each valve chest. The heads are cast separate and contain the ports. Provision is made for slow running and warming up of the engines by means of by-pass to H.P. valve chest and live steam connections with the I.P. and L.P. receivers. The main valves are of the single-ported piston type, there being for each engine, one for the H.P. and two for each I.P. and L.P. cylinders. The main valves have no balance pistons, and the receiver pipes form the valve-chest covers. The valve gear is of the side-shaft type, the valves being actuated by a side shaft, which in turn derives its motion from the main crankshaft by means of gears. The cut-off is fixed at .7 of the stroke at top and .6 of the stroke

at bottom, for the I.P. cylinders; for the H.P., .78 top, .7 bottom; for L.P., .55 top, .44 bottom.

The H.P. piston is of cast steel and the I.P. and L.P. pistons of nickel-steel forgings, dished and fitted with one packing ring, 1 inch wide and ½ inch thick, cut in three segments. The crossheads are forged on lower end of piston rods, and each has a composition slipper, the sliding faces of which are fitted with white metal. The crosshead guide to take the thrust when going ahead is of cast iron bolted to the strongbacks, and is cast hollow for the circulation of water. Carbon-steel gibs are bolted to each side of the go-ahead guides to take the thrust when backing. The side shaft is made of high-grade nickel-steel forgings in four parts, having one crank for the H.P. and two for each I.P. and L.P., and connecting rods of forged steel, fitted with composition boxes lined with white metal. The valve stems have crossheads of bearing metal which take hold of the wrist pin on the connecting rods.

The reversing gear consists of a steam cylinder and a hydraulic controlling cylinder acting on a spiral sleeve on the valve side shaft. The piston rod passes through the hydraulic cylinder. The forward section of the valve shaft has three spirals cut in same. This is fitted with a steel sleeve having spirals running in the opposite direction on the outside, and lined with babbitt on the inside. This movable sleeve is in turn fitted with a composition sleeve fitted with thrust collars on the outside, running in a cast-steel bearing lined with white metal. The movable sleeve has two collars at after end fitted with a ram, having its bearing faces babbitted, which is actuated by two levers set at right angles and keyed on to a steel axle held in bronze bearing bolted to the fore-and-aft angle plates of the main-engine bedplate. The lower of these levers is connected to the crosshead on the end of the reversing-engine piston rod by means of two levers or links. The valve of the steam cylinder is worked by a system of differential levers, the primary motion being derived from the hand lever on the working platform, and the secondary motion from a pin on the crosshead of the reversing engine, all parts being adjusted so that the reversing engine

follows the motion of the hand lever and is firmly held when stopped. The two ends of the oil cylinder are connected by a passage cast in the cylinder having a cock placed in the center, there being an index to show whether the cock is open or shut.

The engines are turned by means of a ratchet and gear operating on a cast-steel worm gear bolted between the couplings of the crank and thrust shafts. There is no steam or electric turning gear.

Each engine is fitted with an 8-inch balanced throttle valve.

There are three oil tanks of 300 gallons total capacity situated on the after bulkhead of the forward engine room, with pipes and locked cocks leading to after engine room.

ENGINE DATA.

Cylinders, number for each engine	4
diameter of H.P., inches	22
I.P., inches	31
L.P., inches	34
Stroke of pistons, inches	20
Valves, diameter of H.P. (one for each engine), inches	11
I.P. (two for each engine), inches	11
L.P. (four for each engine), inches	11
Valve stems, H.P. (1), diameter, inches	1¾
through valve, inches	1½
I.P. (2), diameter, inches	1¾
through valve, inches	1½
L.P. (4), diameter, inches	1¾
through valve, inches	1½
Main steam pipe, diameter, inches	8
Exhaust pipes to I.P. cylinder (2), inches	12
L.P. cylinder (2), inches	14
condenser (2), inches	16
Volume swept by H.P. piston, per stroke, cubic feet	4.32
I.P. piston, per stroke, cubic feet	8.65
L.P. piston (2), per stroke, cubic feet	20.85
Ratio of net area of H.P. to I.P. pistons	2.0
I.P. to L.P. pistons	2.4
H.P. to L.P. pistons	4.83
Clearance of H.P. cylinder, per cent. (aver.)....top, 22.02; bottom....	24.19
I.P. cylinder, per cent. (aver.)....top, 18.36; bottom....	19.88
L.P. cylinder, per cent. (aver.)....top, 15.89; bottom....	17.34
Piston rods, diameter, inches	4¼
length from piston to crosshead, feet and inches	3- 5

Piston rods, axial hole, H.P., diameter, inches	1¾
I.P., diameter, inches	2½
L.P., diameter, inches	2¾
Connecting rods, length from center to center, inches	50
diameter of upper end, inches	4¼
lower end, inches	4¾
axial hole, diameter, inches	2½
crosshead bolts (2), diameter, inches	2½
crank pin-bolts (2), diameter, inches	2½
axial hole, crosshead and crank-pin bolts, diameter, inches	1¼
crosshead pin, length, inches	8½
diameter, inches	5¼
axial hole*, diameter, inch	¾
Crossheads, surface, (ahead), square inches	250
(backing), square inches	67.5
Reversing gear, steam cylinder, diameter, inches	6
stroke, inches	6½

* The crosshead pin consists of a hardened steel bushing, flattened on one side, keyed on to a 2½-inch nickel-steel bolt passing through the forks of the connecting rod, which bolt has a ¾-inch axial hole for purpose of lubrication. Between this bushing and bolt are two recesses, each 2¾ inches long and 1 inch deep, forming a receptacle for oil.

SHAFTING AND BEARINGS.

The crank, thrust, line and propeller shafting is hollow and of forged steel. The crank shaft of each engine is made in two sections, and the shaft of one engine will fit the other. The starboard thrust shaft is supported at its middle by a cast-iron bearing, located abreast the condenser in the after engine room. The propeller shafts are covered with composition at the bearings, as are also the line shafts. The inboard clamp coupling of the line and thrust shafts consists of a steel casting in two sections, 28½ inches long, keyed on shaft by two ⅞-inch square keys, and clamped together by 12 forged-steel bolts.

Each thrust bearing is of cast steel, cast hollow for the circulation of water, and has thirteen steel rings faced with white metal. The bearing is in two pieces, the upper one having an oil box for lubricating the rings, there being a pipe leading to each ring. At each end of the bearing there is a divided stuffing box and gland to prevent the escape of oil. The bearing is supported on two threaded thrust rods of forged steel fitting into sockets in the after bridge of the main-engine bedplate, and

also by cast-iron brackets, bolted to fore and-aft-angle bars to which engine bedplates are bolted, and having slotted holes to permit of fore-and-aft adjustment of the bearing.

Crank shafts, diameter, inches	8
axial hole, inches	4
coupling disc, inches	14
thickness, inches	1¾
coupling bolts in one flange (8), diameter, inches	1½
journals, diameter, inches	8
length, inches	10½, 10½, two of 12¼, 19, 15
length of forward section, feet and inches	6- 6⅝
after section, feet and inches	7- 8¼
Crank pins, diameter, inches	8
length, inches	13
axial hole, diameter, inches	4
Crank webs, width, inches	10
H.P. thickness, inches	3¾
I.P. thickness, inches	4
1st L.P., thickness, inches	4⅜
2d L.P., thickness, inches	4¾
Thrust shafts, diameter, inches	7¾
of coupling discs, inches	15½
axial hole, inches	4½
collars, number, each shaft	10
diameter, inches	11½
thickness, inches	1
distance between, inches	1½
surface, total for both engines, square inches	2,089.16
length, port, feet and inches	4- 6
starboard, feet and inches	20- 5½
Line shafts, diameter, inches	7¾
axial hole, inches	4½
coupling discs, inches	15
length, port, feet and inches	39- 8½
starboard, feet and inches	43- 8½
Propeller shafts, diameter, inches	7¾
axial hole, inches	4½
coupling discs, inches	15
length, feet	24
Intermediate bracket bearings, length, inches	20
diameter, inches	8¼
Stern bracket bearings, length, inches	29½
diameter, inches	8¼
Stern tube bearings, length, forward, inches	30
aft, inches	24

MAIN CONDENSERS.

There are two main condensers, one for each engine. The shells are of rolled copper, stiffened by composition rings, longitudinal seams brazed and lateral joints flanged and riveted. The water chests are also of sheet copper. The tube sheets and supporting plate are of rolled Muntz metal. The circulating water passes through the tubes. Baffle plates are fitted to direct the steam over the tubes. A supporting plate of rolled Muntz metal $\frac{1}{4}$ inch thick supports the tubes at the middle of their length. Water for circulation through condenser is supplied by a centrifugal pump and also by a scoop when the vessel is under way.

Diameter of shell (inside), feet and inches	4–2
Thickness of shell, inch (No. 10 B.W.G.)	.134
Tubes, diameter, inch	$\frac{5}{8}$
Length over heads, feet and inches	9–7$\frac{1}{2}$
length between tube sheets, feet and inches	9–6
thickness, No. 20 B.W.G., inch	.035
Number in each condenser	2,166
Cooling surface, each condenser, square inches	3,359.5
total, square feet	6,719
Ratio of total cooling surface to total heating surface	2.7

MAIN AIR PUMP.

For each main engine there is an independent, three-cylinder, single-acting air pump, driven by pinion and gear from a double engine directly over the air pump. The pump valves consist of single disc of rolled manganese-bronze.

Diameter of steam cylinders (2), inches	4
Stroke, inches	4
Diameter of piston rods, inch	$\frac{5}{8}$
Diameter of pump cylinders, inches	12
Stroke, inches	6
Crankshaft, diameter, inches	2$\frac{1}{2}$
pins, diameter, inches	2$\frac{1}{2}$
Connecting rods, dimensions, inches	1 × $\frac{5}{8}$
Trunk, diameter, inches	4
Kind and diameter of pump valves (manganese-bronze), inches	3
Ratio of volume swept by L.P. pistons per stroke to that of the three air-pump buckets per stroke	17.84 to 1

MAIN CIRCULATING PUMPS.

For each main condenser there is a centrifugal, double-inlet circulating pump, which is arranged to draw from the sea, bilge and main drain pipe, and to discharge into the condenser. It is driven by an independent, vertical, inverted, single-cylinder engine. Each circulating pump is capable of discharging 2,000 gallons of water per minute from the bilge.

Diameter of steam cylinder, inches	4½
Stroke, inches	4
Diameter of pump runner, inches	20
Width of runner, at hub, inches	6
rim, inches	1½
Diameter of inlet nozzle, inches	7
outlet nozzle, inches	7

SCREW PROPELLERS.

The propellers are of manganese-bronze, each with three blades. Each propeller is secured to the shaft by two square keys and a composition nut, screwed on and locked in place. The end of the hub is covered with a composition cap.

PROPELLER DATA.—"LAWRENCE."

Number of blades	3
Diameter, starboard, feet and inches	7-5
port, feet and inches	7-5½
of hub, forward end, feet and inches	1-6½
after end, feet and inches	1-3½
Length of hub, feet and inches	1-4½
Pitch, mean effective, starboard, inches	114.3
port, inches	113.1
Greatest width of blade, on circle, inches	36
Helicoidal area of each screw, square feet	21.2
Projected area of each screw, square feet	16.9
Disc area, square feet	42.6
Pitch ÷ diameter	1.295
Helicoidal area ÷ disc area	.50
Projected area ÷ disc area	.40
Disc area of both screws ÷ I.M.S., at 6 feet W. L.	.804
Immersion of center at mean draught (6½ feet), feet	4½
Center above lowest point of keel, feet	1½
from center line of vessel, feet and inches	3-9

PROPELLER DATA.—"MACDONOUGH."

Number of blades	3
Diameter, starboard, feet and inches,	7-4½
port, feet and inches	7-4⅝
of hub, forward end, feet and inches	1-6½
after end, feet and inches	1-3½
Length of hub, feet and inches	1-4½
Pitch, mean effective, starboard, inches	113.9
port, inches	114.3
Greatest width of blade, on circle, inches	36
Helicoidal area of each screw, square feet	20.7
Projected area of each screw, square feet	17.1
Disc area, square feet	42.4
Pitch ÷ diameter	1.293
Helicoidal area ÷ disc area	.49
Projected area ÷ disc area	.40
Disc area of both screws ÷ I.M.S., at 6 feet W. L.	.804
Immersion of center at mean draught (6½ feet), feet	4⅝
Center above lowest point of keel, feet	1½
from center line of vessel, feet and inches	3-9

BOILERS.

There are, in each vessel, four water-tube boilers of the improved Normand type, all of steel, having one steam drum and three water or mud drums, consisting of steel plates welded longitudinally and having heads welded in. Each boiler has two furnaces, separated by a wall of tubes extending from steam drum to the center mud drum. The grate bars are of mild steel, rolled bars. The furnaces are walled with fire brick at back and front and on sides to height of mud drums. Each boiler rests on saddles which are built up from the hull framing, and are also secured by steel straps passing around mud drums and bolted to the saddles. Each boiler has a twin safety valve 3½ inches in diameter, placed on pad on top of steam drum at the center of its length. The boiler tubes are of seamless-drawn mild steel. The boilers are placed in two watertight compartments, two in a compartment. There are four smoke pipes, one for each boiler, located on the fore-and-aft center line of the ship.

BOILER DATA.

Steam pressure, designed, pounds	250
Number of boilers	4
Length over all, feet and inches	10- 6

Diameter, outside, steam drum, inches	43 9/16
mud drums, inches	19
Thickness of steam drum, top, inch	9/16
bottom (tube sheet), inch	1
mud drums, inch	1/2
Furnaces, internal diameter, forward boiler, feet and inches	4- 4½
after boilers, ft. and ins.	4-10½
length of grate, feet	8
number in each boiler	2
Tubes, outside diameter, inches	1¾
thickness, B.W.G., inches	.11 or .12
number in each boiler	1,980
spaced vertically, inches	1⅞
horizontally, inches	2.28
Heating surface, tube, each boiler (three after), square feet	4,653
forward boiler, square feet	4,014
Grate surface, each boiler (three after), square feet	78
(forward), square feet	70
Smoke pipes (four elliptical), feet and inches	3-1×4
area of all, square feet	41.37
height above lowest grate, feet	23.75
Number and diameter of safety valves for each boiler (two on one base), inches	3½
Diameter of boiler main stop valves, inches	6
auxiliary stop valves, inches	2½
Totals for four boilers:	
Heating surface, tube, square feet	17,973
Grate surface, square feet	304
Volume of steam room, cubic feet	236
Area of water surface, square feet	145
Ratios:	
Tube-heating surface to grate surface	59 to 1
Steam room per square foot of grate surface	1 to 1.3

FORCED DRAFT.

The closed fire-room system of forced draft is used. The air is supplied by four blowers, two in each fire room. The fans are driven by a single-cylinder, vertical, simple, enclosed engine.

Diameter of cylinder, inches	4½
Stroke, inches	4
Diameter of fan, inches	48
Width of fan at rim, inches	14
Area of induction nozzle (each blower), square inches	1,231.5
eduction nozzle (each blower), square inches	2,111

NOTE.—The cowl opening is 3 feet by 3 feet 7 inches; the hole through the deck is 3 feet by 3 feet, and the area of the actual opening is probably less.

FEED PUMPS.

There are two main and four auxiliary duplex feed pumps built from special designs made by the Fore River Engine Co. The main feed pumps are 10 inches by 6 inches by 12 inches, and the auxiliary feed pumps are 5½ inches by 3¼ inches by 6 inches. The main feed pumps are capable of delivering 295 gallons per minute, and the auxiliary feed pumps capable of delivering 72 gallons per minute, each.

The main feed pumps are located one in each engine room, and draw from the feed tanks, the reserve feed tanks, and the air pump channelways, and deliver into the main feed pipe, the auxiliary feed pipe, and into the feed tanks, there being a branch with valves from the feed main to each feed tank.

The auxiliary feed pumps are located in the fire rooms, and draw from the feed tanks and the reserve feed tanks, the boilers in their own compartment, and the sea, and discharge into the auxiliary feed pipe, the fire main, the ash ejectors, and overboard through the bottom blow.

FIRE AND BILGE PUMPS.

In each engine room there is a vertical, simple pump, built from special design of the Fore River Engine Co., having a steam cylinder 6 inches diameter, a water cylinder of 8 inches and a stroke of 12 inches. These pumps draw from the sea and from the bilge, through the auxiliary drain pipe, and discharge into the fire main, overboard, and into pipes for putting out fires in boilers.

WATER-SERVICE PUMPS.

There is no pump for this specific purpose. Water for the water service in engine rooms is taken from a 2-inch nozzle on the inlet elbow of the injection pipe. In addition to this the distiller circulating pump in after engine room may be used for circulating water through the water-service pipes. This pump is a single cylinder, vertical pump of the Davidson type, having a steam cylinder of 3½ inches diameter, a water cylinder of 4½ inches diameter and a stroke of 4 inches.

FEED TANKS.

There is, in each vessel, a feed tank of 150 gallons capacity in each engine room. A part of the tank is fitted as a filter into which the water from the air pump is delivered, the water descending through the filtering material and discharging through the bottom into the main portion of the tank. Each tank has a bolted cover with filter box attached, making it possible to remove cover and filter complete when it becomes necessary to clean the tank. Each tank has a handhole with bolted cover. Each tank has a glass water gauge.

Each tank has the following pipe connections: Discharge from main air pump; overflow pipe to bilge, so arranged that any water passing down it may be seen; suction pipe to feed pumps, with valve; drain pipe from traps; vapor pipe.

Each feed-pump suction is provided with a balanced valve operated by a copper float in the feed tank, so arranged that it will allow no air to enter the feed pipes.

STEAM TRAPS.

The separators, auxiliary steam pipes, the radiators and all places where condensed steam can accumulate are fitted with drain pipes and cocks or valves, with automatic traps, which discharge into the feed tanks. The traps are provided with by-pass pipes and valves for convenience in overhauling.

ASH HOISTS.

Hand rigging only.

ASH EJECTORS.

There is a Davidson 3-inch ash ejector in each fire room, discharging overboard above the water line.

DISTILLING APPARATUS.

In the after engine room is a Davidson stright-tube distiller and two Davidson evaporators. The distiller has a capacity of 955 gallons of potable water per 24 hours (test conditions), at a temperature of 72 degrees Fahrenheit. The two evaporators have a combined capacity (determined by test) of 5,188 gallons of water per 24 hours.

STEERING ENGINE.

In the forward conning tower there is a combined hand-and-steam steering gear. The engine is a simple, two-cylinder engine, having steam cylinders 4 inches diameter and a stroke of 4 inches, designed and built by the Fore River Engine Co. The engine is fitted with a differential valve. The engine is connected to wire leads, the after ends of which are connected to a quadrant, which is keyed directly to the rudder stock. With the steam steering gear the rudder can be put from hard over to hard over in ten seconds when the ship is going full speed. The engine is controlled by hand from the conning tower or from the bridge, and the boat can be steered by hand from either the forward or after conning tower or from the bridge.

STEAM WINDLASS.

There is a steam windlass, built by the Fore River Engine Co., situated on the forward platform deck. The engine is a double-cylinder, vertical, direct-acting one, with cylinders 4 inches diameter and a stroke of 4 inches.

AIR COMPRESSOR.

There is one air compressor (Kaselowsky type) for torpedoes, situated in the after engine room. The compressor has one steam cylinder, diameter, 9.252 inches; stroke, 6.299 inches. The compressor is capable of making 275 revolutions per minute with 42 pounds of steam, the air pressure in the accumulator being 1,500 pounds to the inch. The capacity per hour is 11.649 cubic feet.

TELEGRAPH AND REVOLUTION INDICATORS.

There is a mechanical telegraph in each engine and fire room, built by the Fore River Engine Co., which is connected to transmitters in the conning towers, bridge and engine rooms.

Mechanical telltales are fitted on the bridge and in the forward and after conning towers to show the direction of the revolutions of the main engines. In each engine room there is a mechanical indicator to show the relative speed and direction of revolution of the main engines. The two indicators are connected, and each

has two hands, a red one which turns with the port engine and a green one which turns with the starboard engine. The hands can be turned on their shafts like the hands of a watch, so that by setting them together the relative speeds of the propellers can be seen as the hands separate or remain together in their revolutions over the dial of the indicator.

VOICE PIPES.

For communication between the various parts of the ship there is a system of voice pipes and call bells with annunciators.

ELECTRIC PLANT.

The installation consists in general of one 5-kilowatt generating set, one hundred incandescent lights, one searchlight, one main switchboard, together with the necessary wiring, wiring accessories, molding and fixtures. The generator was made by the General Electric Co. The engine driving the dynamo is a simple, vertical, inverted, direct-acting engine, having a steam cylinder 4.15 inches diameter and a stroke of 4 inches. The bedplate is common to both engine and dynamo. The engine is designed to run at full load at 700 revolutions per minute with 100 pounds steam pressure.

The vessel is wired on the two-wire system.

PRELIMINARY RUNS OF *LAWRENCE*.

The contractors held thirty-six preliminary runs. In all of these runs when the speed of 25 or 26 knots was reached excessive vibrations occurred, which were particularly noticeable in the starboard or forward engine, and were severely felt in the reverse sleeve on the valve shaft.

The gearing driving the valve shaft tends to lock this sleeve in position, and as higher speeds are reached the more firmly it should be held in place. At the critical speeds, it was found to be impossible to hold this sleeve, different parts of the reversing gear breaking at various times.

The following remedies for decreasing vibrations were tried without success: Strengthening the forward part of the engine

foundation; adding counter weights to valve stems; careful adjustment and alignment of crank-shaft and valve-shaft bearings, and babbitting the reverse sleeve.

The following experiments were made to locate the trouble: The main engine was run disconnected from the propeller. (Table A gives H.P. necessary at various speeds.) Running the valve shafts and valves by an electric motor at high speeds. (Table B.) In neither of these experiments was the vibration noticed at the critical speeds in the parts that gave trouble when the engine was run coupled to the propellers.

TABLE A.—FRICTION TEST, STARBOARD ENGINE, TORPEDO-BOAT DESTROYER, NO. 8, *LAWRENCE*, JULY 31, 1902.

R.P.M.	Cylinders, horsepower.				
	H.P.	I.P.	1st L.P.	2d L.P.	Total.
305	51.3	44.3	5.5	5.5	106.6
358	60.3	52.2	6.5	6.5	125.5
185	—12.7	—13.5	65.5	73	112.2*
405	68.7	59	11.7	11	150.2

Shafting aft of thrust shaft was disconnected.
* Throttle closed. L.P. by-pass open.

TABLE B.—RESULTS OF TESTS MADE TO DETERMINE HORSEPOWER REQUIRED TO DRIVE SIDE SHAFT OF STARBOARD ENGINE OF TORPEDO-BOAT DESTROYER NO. 8, *LAWRENCE*, JULY 28, 1902.

Side shaft.	Motor.			
R.P.M.	H.P.	R.P.M.	Volts.	Ampères.
78	2.95	120	38	58
153	5.4	235	61	66
210	8.7	322	95	68
268	11	412	120	68
386	18.8	594	180	78

Pulley on motor, 11.7 inches diameter.
Pulley on engine, 18 inches diameter.

At the suggestion of Rear Admiral Melville special considera- was given the question of torsional vibration of the shafting. The services of Mr. J. H. Macalpine, an authority on high-speed

vibration, was secured. Under Mr. Macalpine's direction the following changes were made: The cut-off in the H.P. cylinder was increased, that of the L.P. decreased. Weights were added to the H.P. and I.P. engines, as follows:

Weights, in pounds.	Starboard engine.	Port engine.
H.P. piston rods, steel plugs,	20	20
Crosshead bolts, lead,	10	10
Connection rods, steel plug,	6	6
I P. piston rod, steel plug,	42	38
Crosshead bolts, lead,	10	10
Connection rod, steel plug,	6	6

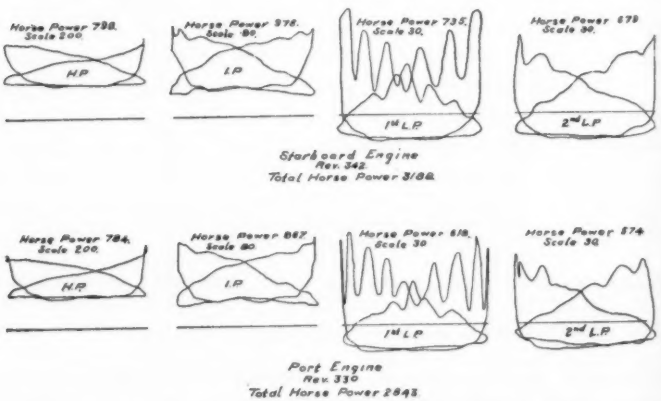
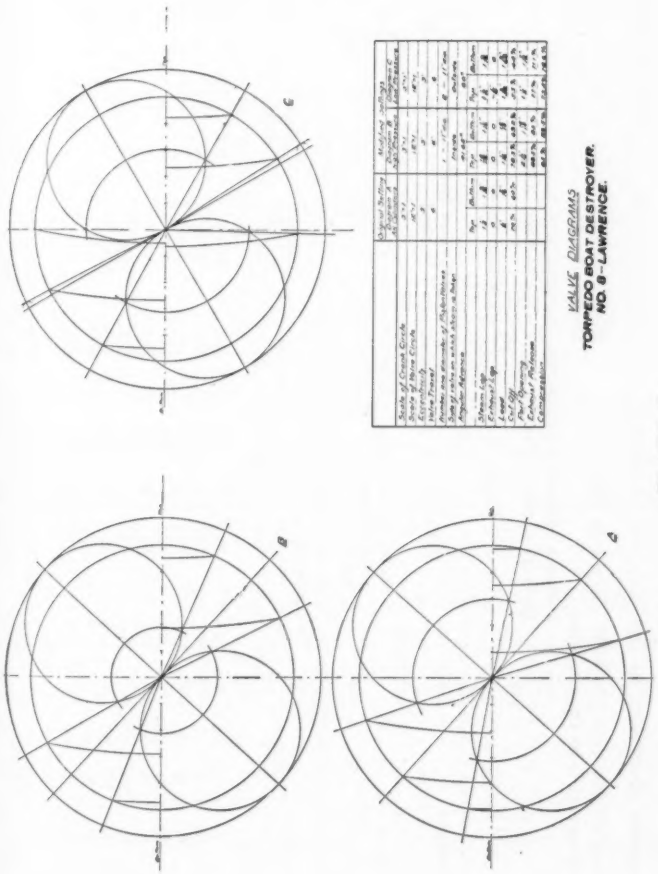


PLATE. 1.—U. S. S. "LAWRENCE," TORPEDO-BOAT DESTROYER. INDICATOR CARDS OF SEPTEMBER 9, 1902.

The effect of these changes was to give a more uniform crank-turning effort. Two trial runs were made without mishap, during which the accompanying cards were taken. The vibration in the reverse sleeve was reduced, but did not disappear.

It is quite evident that this form of valve gear is undesirable for high-speed engines of more than 3,000 horsepower. Any torsional vibration set up in the main shafting is immediately communicated to the valve shaft, and the steam distribution to

the cylinders is changed, which affects the crank-turning effort, and the trouble is thus augmented. The original and modified settings of valves are indicated by the valve diagrams shown in Plate 2.



VALVE DIAGRAMS
TORPEDO BOAT DESTROYER.
NO. 6 "LAWRENCE."

PLATE 2.

TABLE C.—REPORT OF OFFICIAL RUN OF TORPEDO-BOAT DESTROYER NO. 8, *LAWRENCE*, OCTOBER 10, 1902.

Left dock at Quincy, at 7·48 A. M., October 9, 1902.
One run was made to Provincetown.
Nine runs were made over a course of one nautical mile.
Returned to Quincy, at 6·45 P. M., October 10, 1902.

Distance made on previous runs, 2,403 nautical miles.
Distance made on this run..... 152 nautical miles.

Total..... 2,555 nautical miles.

Draught.	Start.	Draught.	Beginning of trial runs.	End of trial runs.	Beginning of hour's run.	Change of trim at 27.735 knots, 4.68 ft. in 246 ft. 3 ins.
Stem.....	5 ft. 3½ ins.	Stem, mean.....	5 ft. 4 5/16 ins.	5 ft. 2½ ins.	5 ft. 7 ins.	
Midship, st'b'd.....	6 6	Stern, mean.....	7 11 9/16	7 10¾	7 10¼	
port.....	6 3½	Mean.....	6 8⅛	6 6½	6 8⅝	
Stern.....	7 11					

Run No.	Direction.	Time.	Revolutions. St'b'd.	Revolutions. Port.	R.P.M. St'b'd.	R.P.M. Port.	Average R.P.M.	Elapsed time.	Speed, knots.	Slip, per cent.
1.....	S.	9 h. 05 m.	748	751	284	285	284.5	2 m. 38 s.	22.785	14.6
2.....	N.	9 13	728	737	285	289	287	2 33	23.529	12.62
3.....	S.	9 25	771	744	330	319	324.5	2 20.2	25.678	15.51
4.....	N.	9 33	747	724	325	315	320	2 18	26.087	12.98
5.....	S.	10 40	763	737	347	335	341	2 12	27.273	14.66
6.....	N.	10 46	764	736	345	332	338.5	2 13	27.068	14.66
7.....	S.	11 33	758	734	370	359	364.5	2 2.8	29.318	14.06
8.....	N.	1 32	768	737	355	341	348	2 9.8	27.735	14.95
9.....	S.	1 40	751	726	364	352	358	2 3.8	29.079	13.33

REMARKS.—Third set of propellers, true screw pitch, 9 feet 6 inches.

TABLE D.—OFFICIAL RUN U. S. S. *LAWRENCE*, OCTOBER 10, 1902.

Time.	Steam.	H.P. chest.	I.P. receiver.	L.P. receiver.	Vacuum.	Air pump R.P.M.	Main feed R.P.M.	Condensing water. Injection.	Condensing water. Discharge.
			FORWARD ENGINE ROOM.						

TABLE D.—OFFICIAL RUN U. S. S. *LAWRENCE*, OCTOBER 10, 1902.

Time.			Steam.	H.P. chest.	I.P. receiver.	L.P. receiver.	Vacuum.	Air pump R.P.M.	Main feed R.P.M.	Condensing water.	
										Injection.	Discharge.
					FORWARD ENGINE ROOM.						
h.	*m.*	*s.*				*lbs.*				*degrees.*	*degrees.*
9	5		125	125	55	7	...	...	...	...	...
9	7	38	125	125	55	7	27	114	16	59	98
9	13		125	125	55	8	...	...	...	...	...
9	15	33	...	125	55	8	26.75	114	20	60	99
9	25		175	175	70	18	...	...	...	...	...
9	27	20.2	175	175	70	18	25.25	124	23	59	108
9	33		170	170	70	19	...	...	...	...	...
9	35	18	175	175	70	19	25.5	120	22	59	106
10	40		200	200	95	25	...	...	...	...	...
10	42	12	200	200	95	25	25	134	26	60	109
10	46		200	200	95	25	...	...	...	...	...
10	48	13	200	200	95	25	24.75	100	20	60	105
11	33		230	230	100	33	...	...	...	...	...
11	35	2.8	...	...	...	33	24	114	26	60	108
1	32		215	215	100	30	...	...	...	...	...
1	34	9.8	225	225	100	30	24	136	34	60	106
1	40		230	230	102	33	...	...	...	...	...
1	42	3.8	230	230	102	33	23.5	132	36	60	100
					AFTER ENGINE ROOM.						
9	5		125	120	50	9	27.5	...	...	...	...
9	7	38	125	120	50	9	27.5	126	22	62	95
9	13		125	120	50	10	27.5	...	...	...	...
9	15	33	125	120	50	10	27.5	126	16	62	98
9	25		175	170	75	16	27	...	...	...	...
9	27	20.2	175	170	75	16	27.25	134	24	64	97
9	33		175	170	75	16	27	...	...	...	...
9	35	18	175	170	75	16	27	132	24	62	98
10	40		200	195	90	22	26.5	...	...	...	...
10	42	12	200	195	90	22	26.5	140	26	64	100
10	46		205	200	90	22	26.5	...	...	...	...
10	48	13	205	200	90	22	26	126	21	63	94
11	33		225	225	100	27	25.5	...	...	...	...
11	35	2.8	210	210	95	23	26	136	25	63	104
1	32		220	215	95	23	24.25	...	...	...	...
1	34	9.8	220	215	95	23	26	138	24	63	102
1	40		225	225	100	25	25.75	...	...	...	...
1	42	3.8	225	225	100	25	26	148	28	64	104

REMARKS.—At 9.45 gauge glass connection on separator in after engine room cracked. At 11.36 operating pipe broke near separator.

The standardization trials took place off Provincetown, Mass., on October 10, 1902, the results of which are shown in Tables C and D. In attempting to run off the one-hour trial at 26 knots, the air-pump frame broke at the rod boss.

In January, 1903, the endurance run of one hour at the required revolutions, 325, was made successfully. (See Table E.) The horsepower-revolution curve is shown by diagram of Plate 3.

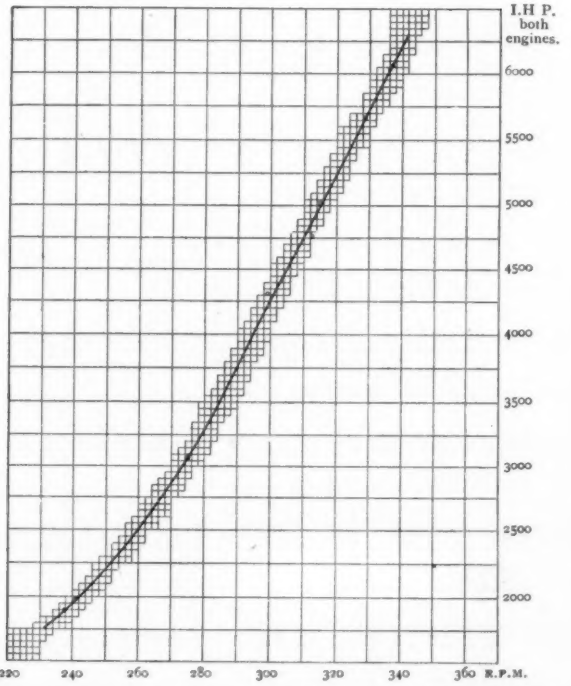


PLATE 3.—HORSEPOWER REVOLUTION CURVE OF MAIN ENGINES OF U. S. S. "LAWRENCE" DURING THE ONE-HOUR ENDURANCE TRIAL, JANUARY, 1903.

New beds for air-pump engines and air-pump crank shaft had been made and fitted of composition instead of cast iron. After her successful trial the boat was fitted with bilge keels.

TABLE E.—FULL-POWER CONTRACT-SPEED TRIAL OF UNITED STATES TORPEDO-BOAT *LAWRENCE*, IN MASSACHUSETTS BAY, JANUARY 7, 1903.

26 knots = 325 revolutions.

Time.	Counter.	Revolutions per minute.	Steam pressure above atmosphere.			Vacuum.	Temperature.			Air pressure.—A. F. R.
			H.P. valve chest.	First receiver.	Second receiver.		Injection.	Discharge.	Feed tanks.	
			PORT ENGINE.							
h. m.			*lbs.*	*lbs.*	*lbs.*	*ins.*				
10·40	610,615	319	200	Gauge disconnected.	18	26	42	87	105	2.7
10·45	612,210	317	...		...	...	...	...	...	2.7
10·50	613,795	318	190		16	26	42	87	110	2.7
10·55	615,385	319	...		...	...	...	...	...	2.7
11·00	616,980	318	195		17	26	41	85	100	2.7
11·05	618,570	318	...		...	...	...	...	...	2.7
11·10	620,160	317	185		15	26	41	85	105	2.7
11·15	621,745	318	...		...	...	...	...	...	2.7
11·20	623,335	319	200		18	26	42	86	110	2.7
11·25	624,930	318	...		...	...	...	...	...	2.7
11·30	626,520	320	190		16	26	42	88	120	2.7
11·35	628,120	319	...		...	...	...	...	...	2.7
11·40	629,716	...	205		19	25.5	42	88	115	2.7
			STARBOARD ENGINE.							F.F.R.
10·40	749,536	337	185	85	25	25	40	103	125	2.7
10·45	751,221	334.4	...	...	...	...	...	...	...	2.7
10·50	752,893	336.8	175	80	25	25	40	90	120	2.7
10·55	754,577	336.8	...	...	...	...	...	...	...	2.7
11·00	756,261	336	180	85	25	25.5	40	95	110	2.7
11·05	757,941	335.2	...	...	...	...	...	...	...	2.7
11·10	759,617	334.4	175	80	25	25.5	40	93	130	2.7
11·15	761,289	337	...	...	...	...	...	...	...	2.7
11·20	762,974	336.6	185	85	25	25	40	95	130	2.7
11·25	764,657	336.8	...	...	...	...	...	...	...	2.7
11·30	766,341	337.6	185	85	25	25	40	95	130	2.7
11·35	768,029	339.8	...	...	...	...	...	...	...	2.7
11·40	769,728	...	185	90	25	25	40	95	130	2.7

OFFICIAL TESTS OF THE *MACDONOUGH*.

The contractors held fifteen preliminary runs with the *Macdonough*. She developed the same defects that were discovered in the *Lawrence*, and the same remedies were adopted.

On January 4, 1903, the boat was standardized off the Provincetown course. The revolution speed curve is shown in com-

parison with that of the *Lawrence* on plate 4. The curves show that the bilge keels of the *Macdonough* give her less speed for the same number of revolutions in the *Lawrence*. The figures for this run are given in Table F.

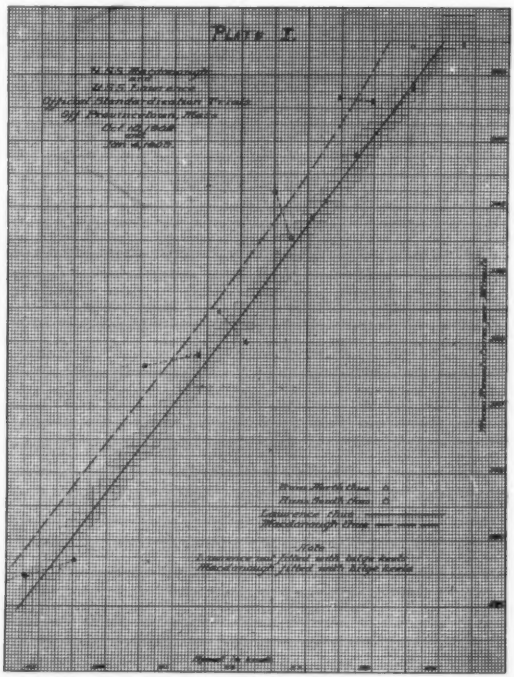


PLATE 4.—SPEED-REVOLUTIONS CURVE, U. S. S. "LAWRENCE" AND "MACDONOUGH."

In the attempt to run off the endurance trial the gear wheels connecting the main shaft with valve shaft carried away, due to excessive vibration in these and adjacent parts.

New gear wheels were fitted, and two more trial runs were held by the contractors. During the first, the highest speed was held only for a few minutes; the vibration in the port reverse sleeves still seemed dangerous.

TABLE F.—OFFICIAL STANDARDIZATION TRIAL OF THE *MACDONOUGH.*
MASSACHUSETTS BAY, JANUARY 4, 1908.

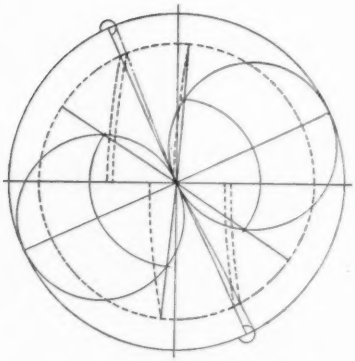
Run.	Speed.	Average revolutions.	Steam pressure above atmosphere.			Vacuum.	Air pressure forward fire room.	Air pressure, after fire room.	Injection.	Discharge.	Feed tanks.	Apparent slip of propellers.
			H.P. valve chest.	First receiver.	Second receiver.							
			STARBOARD ENGINE-ROOM DATA.									
			lbs.	*lbs.*	*lbs.*	*ins.*						
S.	22.75	288.2	124	50.5	9.3	25¾	1¼	1¼	40	70	106	14.85
N.	22.11	288.5	125	50	9.3	25.5	1¼	1¼	40	75	112	16.47
S.	25.37	332.3	173.6	75.6	20.3	26	2	1½	40	74	112	14.95
N.	24.55	331.1	175	75.3	21	25.3	1	2	40	74	113	17.23
S.	26.74	353.4	204	92.6	27	25¾	2¼	2¼	40	80	118	15.05
N.	26.51	362.3	217.6	99	29.6	25¾	3	3	40	89	120	17.51
S.	27.99	375.9	233.3	112.3	34	25	3¾	3¼	40	82	170	16.19
N.	27.48	377.4	236	112.2	34	25.5	3¾	3¾	40	89	173	17.87
S.	28.51	382.8	245	112.6	35	25	3½	3½	40	82	150	16.43
			PORT ENGINE-ROOM DATA.									
S.	22.75	281.4	125	50	10	26	...	...	40	74	85	...
N.	22.11	275.6	125	50	10	26	...	...	40	72	102	...
S.	25.37	303.6	171.6	66.6	12.6	26	...	...	40	72	108	...
N.	24.55	300.4	178.3	71	11.6	26	...	...	40	72	105	...
S.	26.74	317.3	201.6	84.6	16	26	...	...	40	78	110	...
N.	26.51	322.5	216.6	94.3	18.3	26	...	...	40	78	115	...
S.	27.99	330.2	236.6	100	21.3	26	...	...	40	81.3	110	...
N.	27.48	335.7	238.6	101.3	23.6	26	...	...	40	81.6	100	...
S.	28.51	345.7	251.6	110	24.6	26	...	...	40	82	115	...

For the second run the valves on the port L.P. engines were set at an angular advance of 65 degrees, being 5 degrees greater than that of the starboard engine. See valve diagram shown in Plate 5.

This valve setting eased the vibration already mentioned considerably, and the accompanying cards shown in Plate 6 were taken; the figures for the run being shown in Plate G.

The work done on these boats in tuning them up for their official trials shows the technical skill and experience necessary to design the machinery of torpedo boats. Such machinery must be well balanced, properly proportioned, carefully built and efficiently installed.

The importance of efficient auxiliaries is also manifest, while



Valve Readings
ang. adv. 65°

	Top	Bott.
steam lap	1 1/4"	1 1/8"
exh lap	- 1/4"	0"
lead	1/8"	1/8"
cut	18 11/16"	17 1/4"
exh rel.	72.3%	66.6%
comp.	69.2%	75.6%

Note- Other valve readings taken at full stroke

L.P. Engine—Port.

PLATE 5.—VALVE DIAGRAM, U. S. S. "MACDONOUGH."

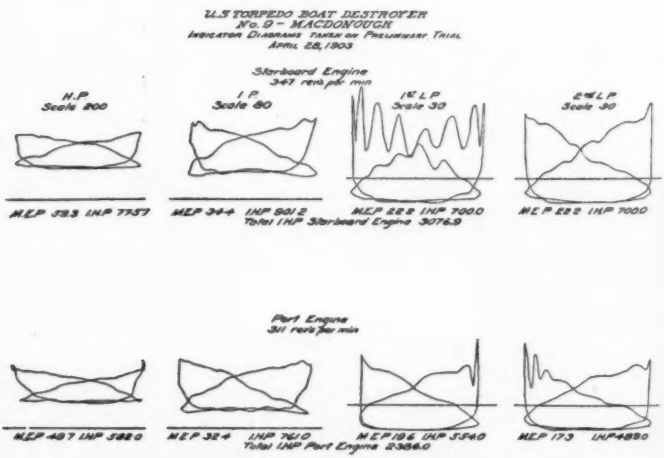


PLATE 6.—"MACDONOUGH" CARDS.

skill and endurance is essential upon the part of those who run the trial trips.

No firm should engage in the construction of such machinery unless it has had extended previous experience in the building of light fast working engines.

TABLE G.—FULL-POWER CONTRACT SPEED TRIAL OF UNITED STATES TORPEDO-BOAT DESTROYER NO. 9, *MACDONOUGH*, IN MASSACHUSETTS BAY, MAY 27, 1903.

26 knots = 331 revolutions.

Time.	Revolutions per minute.	Steam pressure above atmosphere.			Vacuum.	Injection.	Discharge.	Air pressure in fire rooms.
		H.P. valve chest.	First receiver.	Second receiver.				
		PORT ENGINE.						
		lbs.	*lbs.*	*lbs.*	*ins.*			
10'45	313	225	100	25	24	48	100	3
10'55	320	200	85	25	24	50	98	3
11'05	313	220	100	30	24	50	98	3
11'15	319	220	100	27	24	50	98	3
11'25	313	205	95	26	24	50	98	3
11'35	323	225	100	28	24	50	98	3
11'45	317	225	100	28	24	50	98	3
		STARBOARD ENGINE.						
10'45	351	220	100	27	25.5	48	96	...
10'55	356	205	90	27	25.5	50	94	...
11'05	337	220	100	27	25.5	50	95	...
11'15	346	225	100	27	25.5	50	98	...
11'25	342	215	105	25	25.5	50	99	...
11'35	359	240	110	32	25.5	50	98	...
11'45	347	225	100	25	24.8	50	100	...

On May 27, 1903, the endurance trial was successfully held outside of Boston Harbor.

DESCRIPTION AND TRIALS OF THE U. S. TORPEDO-BOAT DESTROYERS *HOPKINS* AND *HULL*.

By John H. Macalpine and Alfred C. Layman, Associates.

The *Hopkins* and *Hull* are twin-screw torpedo-boat destroyers of a specified trial displacement of 408 tons. Their designed draught is about six feet. They were built by the Harlan and Hollingsworth Company, of Wilmington, Delaware, a corporation which is now a branch of the United States Shipbuilding Company. The designs were made by the builders in accordance with the general requirements of the Navy Department.

Contract.—The contract was signed on October 19, 1898, the price being $291,000 for each vessel. This price is exclusive of the articles supplied by the Government; that is, ordnance, ordnance stores, a portion of the equipment, and some other minor outfit.

Trial Conditions.—A speed of 29 nautical miles per hour was guaranteed, to be maintained for two consecutive hours. There was no speed premium. But there was a penalty at the rate of $32,000 per knot for deficiency of speed below 29 knots; the vessel to be rejected should the speed not exceed 28 knots.

The following are the specified trial weights:

Ordnance, tons	15
Coal, tons	25
Equipment, including crew, tons	12
Total, tons	52

All torpedo boats and torpedo-boat destroyers authorized by Congress at the same time as the *Hopkins* and *Hull* had the conditions of the speed trials modified by the Secretary of the Navy acting on the recommendation of the Chiefs of the Bureau of Steam Engineering and of Construction and Repair. The

destroyers were now required to standardize up to the highest attainable speed, which should not fall below 28 nautical miles per hour. An endurance trial was then to be made for one hour at an average speed of at least 26 knots.

HULLS.

The metal of the hulls is principally mild steel of domestic manufacture; but the sheer strake and the top strake of the bunkers on each side are of nickel-steel. The frames are spaced 21 inches apart, except in the engine compartments, where they are reduced to 18 inches.

The keel plate is 13 pounds per square foot for two-thirds of the length amidships, being reduced to 10 forward and aft of this. The vertical keel is 12, the outer plating 9½ for two-thirds of the length amidships, reduced to 7 at the ends. Bilge, bottom, and garboard strakes are 10½, reduced to 8 at the ends.

Watertight Compartments.—The vessels are each divided into 13 watertight compartments, as follows:

1. Paint room.
2. Windlass and stores.
3. Crew's quarters, with storeroom below.
4. Firemen's quarters, with forward magazine below.
5. Machinists' and petty officers' quarters, with storeroom below.
6. Forward boiler room.
7. Forward engine room.
8. After engine room.
9. After boiler room.
10. Cooks' and stewards' quarters, and galley, with storeroom, and 300-gallon water tank below.
11. Officers' quarters and bathroom, with after magazine and 2,000-gallon water tank below.
12. Officers' mess room, dispensary and pantry.
13. Tiller and storeroom.

The inner sides of plating liable to sweat from differences of temperature, as the under side of the deck, sides of ship and some bulkheads, especially in the living quarters, were cork painted.

These quarters were also ceiled under the deck beams with canvas and at the sides with asbestos sheets over wire netting. This was held in place by brass strips, the screws entering furring strips fastened to the beams and frames.

Double Bottom.—For nearly the whole length of the engine and boiler space the vessels have a double bottom varying from 4 feet 3 inches to 8 feet 7 inches broad. The space thus formed is utilized for feed-water tanks, one being placed in each of the four compartments.

Decks and Conning Towers.—The main deck extends completely fore and aft and on it are built the forward and after conning towers. The forward tower is of nickel-steel plate, 20 pounds per square foot; the after of mild steel, 10 pounds per square foot. Each tower carries a gun platform and an extension aft of that on the forward tower is used as a bridge, having a steering wheel, compass, and searchlight. On the main deck are placed the lighter guns, torpedo tubes, various chests, and the boats.

There is a turtle deck forward extending about 45 feet, under which two torpedoes and various other portable fittings are stowed.

The berth deck extends forward and aft except in the machinery space. It is watertight and thus adds to the safety of the ship.

Coal Bunkers.—The engine and boiler compartments have a coal bunker on each side, making in all eight bunkers. Each is watertight. These when full afford very complete side protection to the more vital parts of the ship. The total coal capacity is 154 tons.

Drainage System.—Each circulating pump has a 7-inch bilge connection, the forward pump drawing from the forward engine and boiler compartments, and the after pump from the after engine and boiler compartments.

A main drain pipe extends fore and aft, having connections to all except the end compartments, which drain by sluices. The pipe is 3 inches diameter in the machinery spaces, reduced to 2 inches beyond this. The fire and bilge pump sucks from this pipe.

Boats.—Each vessel is provided with two 20-foot cutters and one 20-foot whale boat.

Armament.—

2	3-inch rapid-fire, 50-caliber.
5	6-pdr. semi-automatic rapid-fire.
10	6-mm. rifles.
72	38-caliber revolvers.
1	45-caliber rifle.
1	22-caliber rifle.
4	mark I, 5-meter torpedoes.
2	central-pivot torpedo tubes.
4	warheads.

Hull Data.—Lengths are in feet, areas in square feet, weights in tons.

The vessel is supposed to be at the designed *loaded* draught of 6 feet 6 inches, except where otherwise noted.

Length over all	248½
on L.W.L.	244
Beam on L.W.L.	24½
Ratio of length to beam	10.03
Depth at side, from top of beams to under side of frames, at midship section	14.46
Draught, mean	6.5
Displacement	465
per inch at L.W.L.	10
Center of gravity of L.W.L. plane, aft of midship section	13.25
buoyancy above bottom of keel	4.22
aft of midship section	4.25
Number of frames	145

Stability Data.—The following data, except the last item, are from the *Hopkins* inclining experiment made on 26th May, 1903, just before delivery to the Government. She was then practically complete except the torpedo tubes, guns, ammunition, and canvas; and there was no coal or water aboard. The figures given here are uncorrected for these articles.

Draught, forward, feet and inches	3–9½
aft, feet and inches	9–9½
mean above base line	6–0½
Displacement, tons	411

The draught figures are for the 8-feet diameter propellers, and in estimating displacement the water of Christiana Creek is taken at 63 pounds per cubic foot.

Transverse metacentric height, feet	4.788
Longitudinal metacentric height, feet	819

The last item is from the calculated curves.

MACHINERY.

Main Engines.—There are two vertical, inverted, four-cylinder, triple-expansion engines, the port engine being in the forward and the starboard engine in the after compartment. The H.P. cylinder is forward and the two L.P.'s aft. The engines are of what is generally known as the Belliss type; that is, one piston valve placed between the H.P. and I.P. cylinders is common to both, and similarly the two L.P. cylinders have one valve. This considerably shortens the engine and requires only two valve gears.

The framing consists of wrought-steel columns, braced; and the bedplates are of cast steel.

The valve gears are of Stephenson type, with double bar links. The reversing gear is fitted both for hand and steam, and the reversing levers are slotted in the usual way to vary the cuts off of each valve independently.

Engine Data.—Dimensions are in inches.

Cylinders, number for each engine	4
Diameter of H.P. cylinder	22
I.P. cylinder	32½
each L.P. cylinder	34
Stroke of pistons	18
Diameter of H.P and I.P. valve	17
L.P. valve	17½
main steam pipe to each engine	9¾
exhaust to condenser	18½
piston rods, outside	4½
connecting rods, at top and bottom	4, 4¾
inside; H.P., I.P. and L.P.	2, 2, 2½
Length of connecting rods, center to center	40
Crosshead pin, diameter by length	4¾ × 8
Main slipper surface, ahead, square inches	187

Main slipper surface, astern, square inches	85
Diameter of valve stems	2¼
Reversing engine, diameter of cylinder	7
stroke	10

The pistons are of cast steel; the piston rods, crossheads, connecting rods, and valve stems of nickel-steel, oil tempered and annealed. The main valves are of cast iron.

Shafting and Bearings.—The main shafting is all hollow, forged of nickel-steel, oil tempered and annealed. Tensile strength, 95,000 pounds; elastic limit, 65,000; elongation, 21 per cent. in two inches, are the minimum requirements.

Each crank shaft is in one piece. The cranks of the H.P. and I.P. cylinders are at 180 degrees as also are those of the two L.P. cylinders; but the latter are at right angles to the former. The eccentrics are of cast iron keyed on.

The thrust blocks consist of a cast-steel base plate into which are set removable top and bottom castings of composition, cored out for water circulation. The collars, which form complete rings, are babbitted, and there are stuffing boxes to retain the oil.

The shafting of the forward engine has two spring bearings forward of the stern tube; and that of the after engine, one. Each shaft is well supported at both ends of the stern tube, and each line has two strut bearings lined with white metal, outside the ship, the shafts having composition sleeves at these outer bearings.

Shafting Data.—All dimensions are in inches, except where otherwise stated.

Crank Shaft.—

Diameter, outside	7¼
inside	4
of crank pins	7¼
Length of crank pins	13
Webs, breadth and thickness	9 × 4½
Diameter of coupling	14½
Thickness of coupling	2½
Six coupling bolts, diameter	1¾
Six journals, length	12
One journal, length	13½
Length of shafts, feet	17

Thrust Shaft.—

Diameter, outside	7¾
inside	4
Collars, number on each shaft	12
diameter	11
thickness	1
space between	1¾
total surface for each engine, square inches	628
Length, port shaft, feet and inches	20–6
starboard shaft, feet and inches	19–4

Line Shafting.—

Diameter, outside	7¼
inside	4
Length, feet and inches	24–4½

This line shafting is in the forward engine only. In the after engine the thrust shaft connects direct to the stern-tube shaft.

Stern-Tube Shafts.—

Diameter, outside	7½
inside	4
Length, feet	30

Propeller Shaft.—

Diameter, outside	7½
inside	4
Length, feet and inches	33–7½

Stern-Tube Bearing.—

Length, forward	21¾
aft	23¾

Strut Bearings.—

Length, forward	24
aft	30

Propellers.—The propellers are shown in Figs. 1 to 4. That in Figs. 1 and 2 is a design bought for these boats from a celebrated firm of torpedo-boat builders abroad. That shown by Figs. 3 and 4 was designed by the present writers. The propellers will be referred to as X, Y, and Z; Y and Z being that given by Figs. 3 and 4 when 8 feet 6 inches and 8 feet diameter, respectively.

Propeller X was cast by the Hyde Windlass Company, Bath, Maine; and Y by the Crown Smelting Company, Chester, Pa. They are both of high-grade bronze, giving very regular tests.

X

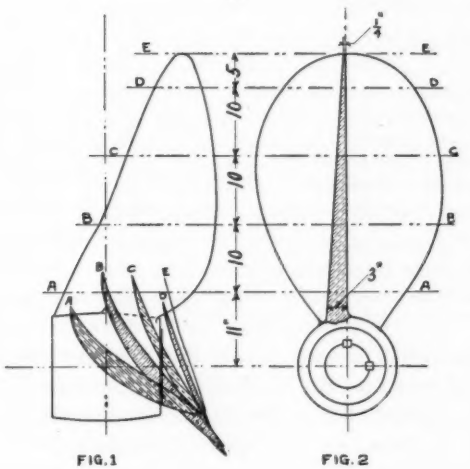


FIG. 1 FIG. 2

Y, 8 FEET 6 INCHES DIAMETER
Z, 8 FEET 0 INCHES "

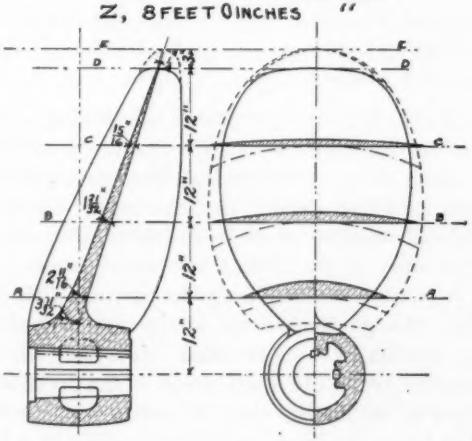


FIG. 3 FIG. 4

Propeller Data.—Lengths are in feet; areas are in square feet; weights are in pounds.

	X.	Y.	Z.
Number of blades	3	3	3
Diameter	7⅞	8.5	8
Mean pitch, port	8.34	8.38	8.38
starboard	8.20	8.51	8.51
Disc area of one propeller	46.16	56.74	50.26
Helicoidal area of one propeller	21.75	25.94	25.11
Projected area of one propeller	17.5	22.0	21.25
Weight of one propeller	2,029.5	2,217	2,203
Tensile test, mean of coupons	59,840	70,660	
Elastic limit, mean of coupons	25,957	31,890	
Elongation in 2 inches, mean of coupons, pr.ct.	36	31.75	
Contraction of area, mean of coupons, pr.ct.	35	29.6	

The mean pitch in each case is that measured from the castings; for X, that from the *Hull's* propellers, and for Y and Z that from the propellers with which *Hopkins* ran the official trial. All propellers decreased in pitch slightly in casting.

In propeller X the pitch is variable both in passing outward radially and also along points at the same radius. Measured from the original drawing, the maximum pitch at the various sections is

A, 9 feet 2 inches.
B, 9 feet 3 inches.
C, 9 feet 4 inches.
D, 8 feet 10½ inches.
E, 7 feet 11 inches.

The pitch also fines as we approach the leading and following edges—not so much as the curvature of the sections appear to show, for these being plane sections they would be somewhat curved even for a true screw. But, calculating the true pitch at the ends of each section, we find a variable diminution averaging about 20 per cent. at the leading and about 10 per cent. at the following edge from the maxima given above. This is quite an old practice, though we are not aware that it had ever been proved by experiment to be of value. It is used here to a very marked amount, and while it will diminish the pressure gradients at the edges we would fear their becoming very great opposite parts of the blade further in, and thus leading to a breakdown

of the efficiency of the propeller at comparatively low mean pressure. So far as one may safely judge from the single comparison, the trials of *Hopkins* and *Hull* seem to justify this fear.

Considering the fining of the edge pitches, it seems to us hardly just to calculate slips from the mean pitch of the whole blade. Thus while for the *Hull's* official trials we use 8.27 pitch, from the official report (the mean of the values given above under *Propeller Data*) we have also given these for 8.5 feet pitch, and probably a slightly greater figure would have been truer.

It was determined, should propeller X not prove satisfactory, to adopt a true screw. The poor performance of X at the high speeds gave unreliable data, hence propeller Y was made large at the first as it could be so readily reduced.

To insure propeller Y being a true screw, the pattern was carefully tested by sweeping and small corrections made. But in casting it fined most toward the tip. Thus the means from the two propellers of *Hopkins*, taking all blades, gave—

At section A, 8.69 feet pitch. At section C, 8.355 feet pitch.
B, 8.463 feet pitch. D, 8.312 feet pitch.

This decrease of pitch towards the tip will probably make more uniform pressure over the blade than for a true screw, in which case breakdown of efficiency may not occur so early, though we are not aware that experiments support this view.

The blades of X and Y are of nearly equal strength at the root. The various sections of Y are equally stressed when the blades are uniformly loaded. This considerably thinned the blade, saving weight and raising efficiency.

Condensers.—There is one cylindrical condenser for each main engine. The shell is made of copper sheets, brazed, and stiffened by brass T rings. The tube sheets are of rolled Muntz metal. The tubes are of Muntz metal, tinned inside and outside, and are supported in the center by a plate, No. 11 B.W.G. thick. They are spaced $\frac{15}{16}$ inch apart, and the circulating water passes through them.

Condenser Data.—Lengths are in inches, surfaces in square feet.

Shell, diameter, inside	45
length between tube sheets	126

Shell, thickness, No. 10 B.W.G.	0.132
Tubes, diameter, outside	$\frac{5}{8}$
thickness, No. 20 B.W.G.	0.036
number in each condenser	1,744
Cooling surface, in each condenser	2,996
Ratio of total cooling to total grate surface	1 : .0491
heating surface	1 : 2.939

Air Pumps.—At the forward end of each main engine a single-acting vertical air pump is placed. It is actuated by a nickel-steel crank shaft of $3\frac{1}{4}$ inches diameter, bolted to the forward end of the main crank shaft; this, by means of a sliding block, actuates a yoke bolted to the top of the bucket trunk.

The head, bucket and foot valves are each 7 in number, the openings being $4\frac{1}{2}$ inches diameter. Each valve consists of two Tobin bronze flat discs $\frac{1}{8\frac{1}{2}}$ inch thick, having $\frac{1}{2}$ inch lift. After some trials the head valves were altered by the addition of conical springs of No. 11 B.W.G. phosphor-bronze wire, as it was suspected that there was considerable leakage back on the top of the bucket causing an abnormal rise of pressure at top stroke. The alteration seemed to relieve the pump, though definite information could not be obtained.

Air Pump Data.—Lengths are in inches.

Diameter of pump	21
Stroke of pump	5
Diameter of trunk	$7\frac{1}{2}$
suction pipe	$6\frac{1}{2}$
discharge	$5\frac{1}{2}$
Ratio of volume of L.P. cylinders to that of air pump, $2 \times 34^2 \times 18 \div 21^2 \times 5 = 1 : 18.87$	

The main feed pump in the engine room is used to keep the condenser free of water when the main engines are not working.

Circulating Pumps.—In each engine room there is a double-inlet, water-balanced, vertical centrifugal pump to supply water to the condensers when the vessel is at rest or moving too slowly to make the scoops efficient. The pump-case and runner are of composition and the shaft of Tobin bronze, running on end-grain lignum vitae. Each pump is driven by a two-cylinder vertical inverted engine.

Circulating Pump Data.—Lengths are in inches.

Steam cylinders, two, diameter	3
stroke	4
Pump runner, diameter	18¾
width at periphery	2 5/16
Inlet and outlet diameters	7

Feed and Filter Tanks.—Each air pump discharges into a feed tank in its engine room of about 200 gallons capacity. Part of each tank is used as a filter, and so arranged that the filtering material will always be fully immersed. The filtering material used by the contractors with excellent results was sponge, but precautions were taken to cover the sponge side of the perforated discharge plate of the filter box with very fine copper gauze, to prevent particles of disintegrated sponge getting through the feed system. To increase the length of service of the sponges, it was found best not to lead any steam drains into the feed tank.

The feed suction is regulated by a balanced valve operated by a copper float. There are also all the usual accessories, such as overflows, lightly-loaded escape valves, etc., etc.

Reserve Feed-Water Tanks.—As stated earlier, the reserve feed-water tanks are formed by the double bottom, one in each engine and boiler compartment. Their capacities are as follows:

After boiler room, tons	9.7
engine room, tons	8.8
Forward engine room, tons	9.2
boiler room, tons	10.4

Main Feed Pumps.—There is one main feed pump of the Blake vertical, simplex type, in each engine room and one in each boiler room. Steam cylinder, 10 inches diameter; water cylinder, 6 inches diameter; stroke, 12 inches. Each feed pump sucks from the corresponding filter tank and also from the main suction line connecting to all reserve feed-water tanks. They discharge into the main or auxiliary feed line, through or past the feed heater.

Auxiliary Feed Pumps.—There is one in each boiler compartment and they are duplicates of the main feed pumps.

In addition to the same suctions as the main feed pumps, each pump also draws from the sea and from the bottoms of boilers in its own compartment. They discharge into the auxiliary feed line, the fire main, the ash ejectors, and overboard through the bottom blow.

Feed-Water Heaters.—There is one in each engine room through which the main feed pump in the same compartment can discharge.

Inside diameter of shell, inches	14
Tubes, diameter, outside, inch	$\frac{3}{4}$
number	151
thickness, No. 14 B.W.G., inch	0.084
length between tube plates, inches	71
surface, square feet	175.6

Boilers.—There are four boilers, two being placed in each boiler compartment. They are of the Thornycroft *Daring* (modified) type. Each boiler has an upper and three lower drums. The upper drum is connected to each of the others by uptake tubes and to the central lower drum by large downtake tubes; this latter is also connected at the back by a large pipe to each of the side drums.

Instead of a single wall formed by the uptake tubes from each side drum, as in the *Daring* type, there are here eight rows of tubes. The tubes of each pair of rows come into line fore and aft to form altogether four walls on each side of the boiler to constrain the flow of the hot gases.

The drums are made of open-hearth steel, with welded longitudinal and circumferential seams. The small tubes are of low-carbon mild steel, seamless drawn, and the downtakes are of lap-welded mild steel.

Each boiler has two furnaces and four furnace doors. The furnaces are lined with fire brick $1\frac{1}{2}$ inches thick in front, back 2 inches, and sides up to the undersides of the tubes, also 2 inches.

The grate bars are of mild steel $\frac{5}{16}$ inch thick at top, $\frac{1}{8}$ inch thick at bottom, by 3 inches deep at center, riveted together in groups of six bars each.

The boilers are lined outside with sheet asbestos $\frac{1}{2}$ inch thick,

which is covered inside and outside by galvanized sheet steel, No. 22, U. S. S. gauge (.031+ inch) in conveniently removable sections.

There are all the usual mountings, besides Thornycroft automatic feed regulators, mica gauges and fire extinguishers.

The main steam pipe from each boiler and the auxiliary steam pipe in each boiler room was fitted with a Foster automatic safety stop valve, closing on sufficient reduction of pressure or by opening an emergency valve from the engine or boiler rooms. These were carefully tested and found to be very efficient.

Boiler Data.—Lengths are in inches, except where marked; surfaces are in square feet.

Test pressure, pounds	360
Safety valves set on trial to, pounds	280
Number of boilers	4
Extreme width over casings, feet and inches	14- 8½
length over steam drum, feet and inches	12- 4
height over steam drum from bottom of ash pan	10- 6½
Steam drum, diameter, inside	36
length, feet and inches	12- 0½
thickness	1⅛
Center water drum, diameter, inside	19
thickness	¾
Wing water drums, diameter, inside	7⅝
thickness	½
Furnaces, total width for one boiler, feet and inches	10- 6
length, feet	7
Tubes for one boiler.	
Uptake tubes, next furnace, number	372
outside diameter	1¾
thickness, No. 11 B.W.G	.120
Other uptake tubes, number	1,346
outside diameter	1⅝
thickness, No. 11 B.W.G	.120
Downtake tubes, number	14
outside diameter	4
thickness	5/16
Horizontal tubes, connecting water drains, number	1
outside diameter	7
thickness	¼
Heating surface for one boiler	4,403

Grate surface for one boiler	73.5
Heating surface for four boilers	17,612
Grate surface for four boilers	294
Ratio heating surface to grate surface	59.9:1
Funnels, number	4
length by breadth (oval), feet and inches	5-11 × 3-1
area of each	13.6
height above grate bars, feet and inches	29-3
Ratio grate surface to funnel area	5.40:1
Safety valves, two to each boiler, diameter	3½
Main stop valves, diameter	6½

Forced Draft.—The closed boiler-room system of forced draft is used. There are two single-inlet blowers in each boiler compartment. These with their engines are of the Sturtevant design. The engines are horizontal, two cylinder; the axis of the crank shaft is vertical and carries the fan at its upper end. The engines are completely inclosed and the lubrication is forced, the lubricant being supplied from and pumped back to cylindrical tanks just under the deck, which are under a few pounds per square inch of air pressure.

Forced-Draft Fan and Engine Data.—Lengths are in inches.

Cylinders, two, diameter	5
stroke	4
Blowers, diameter	54
number of vanes	8
width at rim	14
diameter of inlet	30

Fire and Bilge Pumps.—In each engine room there is a bilge pump by the Geo. F. Blake Mfg. Co., simplex, vertical, the steam cylinder diameter being 6 inches and the water cylinder 8 inches, by 12 inches stroke.

These suck from the sea, the bilge through the 3-inch drain pipe, and dicharge into the fire main, overboard, and to the fire extinguishers.

Engine-Room Water Service.—This is supplied from a 2-inch connection on the inlet water chest of each condenser and one of the same size from the distiller circulating pump.

Steam Traps.—All drain pipes from the ship and machinery,

except those from the cylinders of the main and auxiliary engines and pumps, are led into automatic traps. These traps have by-pass valves for convenience in overhauling.

Steam Ejectors.—There is a steam ejector, of a capacity of 2,000 gallons per hour, in each boiler and engine room. Each sucks from the bilge of its own compartment and discharges overboard above the water line.

Ash Ejectors.—In each boiler compartment there is an ash ejector, the discharge pipe being 5 inches diameter. This works by water from the auxiliary feed pump.

Telegraphs, &c.—There are the usual pressure gauges, revolution counters, telegraphs from bridge and conning towers to engine rooms, between forward engine and boiler rooms and between after engine and boiler rooms; also voice pipes from bridge and conning towers to engine rooms, and from engine to boiler rooms.

There are also metallic thermometers to give the temperature of the main steam, the feed water at various points, and the entering and discharge circulating water.

There is a revolution indicator with a dial in each engine room with two hands, one for each engine. Each hand revolves once for one hundred revolutions of its engine; and as the hands can be set together at any moment the engines may readily be adjusted to run together.

Steering Engine.—There is a steering engine by Williamson Bros. placed under the turtleback just ahead of the forward conning tower. It is horizontal, with two cylinders $4\frac{1}{2}$ inches diameter by $4\frac{1}{2}$ inches stroke, and is capable of putting the rudder from hard over to hard over—that is, through an angle of 70 degrees—in 12 seconds when the ship is at full speed.

It is actuated by means of steering wheels (which can also be changed to hand gear) on the bridge and in the conning towers. From the drum of the engine a $\frac{1}{2}$-inch diameter steel wire rope passes aft and round a sheave attached to a slider. From this slider a rope $\frac{5}{8}$-inch diameter, which is under double the stress of the forward rope, passes aft and over the rudder quadrant.

Capstan and Windlass.—The windlass is placed in the second

forward watertight compartment on the berth deck. It has a worm wheel which actuates independently two drums, each holding 75 fathoms of wire rope of 2¾ inches circumference.

The capstan is on the turtle deck and can be operated by hand or by the windlass engine.

Distilling Apparatus and Evaporators.—There are two evaporators and one distiller with their pumps placed in the after engine room. Each evaporator has a capacity of 1,500 gallons of water in 24 hours, and the distiller a capacity of 800 gallons of potable water at 90 degrees Fahrenheit in 24 hours.

The brine pump for supplying the evaporators with salt water and also for pumping them out is a Blake simplex vertical pump; steam cylinder, 3½ inches; pump, 2⅛ inches; stroke, 4 inches.

The pump for circulating cooling water in the distiller is a Blake simplex; steam cylinder, 3½ inches; pump, 4¾ inches; stroke, 4 inches.

Electric-Lighting Plant.—Electricity is supplied by a 5-kilowatt generating set, consisting of a dynamo and engine, by the General Electric Co., running at 700 revolutions, with steam reduced to 100 pounds per square inch.

The engine is vertical, with one cylinder 4 inches diameter by 4 inches stroke. It has a flywheel governor.

The generator is of the direct-current, compound-wound multipolar type.

The vessel is wired on the two-wire system, in five circuits.

There are in all 108 incandescent lamps of 16-candle power, 6 of 5-candle power, and also the attachments for firing the torpedoes from the tubes. A searchlight is placed on the bridge, above the forward conning tower. It is manufactured by the General Electric Co. The lens is 24 inches diameter and the lamp is arranged for a current of 50 ampères and 80 volts.

Torpedo Air Compressor.—The torpedo air compressor is Type V of the Rand Drill Co. The steam and air cylinders are in duplicate; that is, the machine is symmetrical about a central transverse plane. The compression takes place in three stages, and the machines are built to deliver 12 cubic feet of air per hour under a pressure of 1,700 pounds per square inch.

There are two steam cylinders $5\frac{3}{4}$ inches diameter by $3\frac{3}{4}$ inches stroke.

Air cylinders, two each of $6\frac{3}{8}$, $5\frac{1}{4}$, and $1\frac{3}{32}$ inches diameter by $2\frac{1}{2}$ inches stroke.

Analysis of Weights of Ship Complete.—Without going into great detail, it is difficult to give any useful division of ship weights beyond the following.

Taking the ship without coal, water, or crew aboard we have—

Steel work, including conning towers, but not including such as davits, handrails, etc.	= 44 per cent. of total.
Machinery weights, dry	= 45 per cent. of total.
Battery of guns, torpedo tubes and small arms	= 0.5 per cent. of total.
	89.5 per cent. of total.

This leaves about 11 per cent. for all hoisting and anchor gear, boats, woodwork, drainage system, outfit, etc., etc.

Machinery weights, in tons.—	Dry.	Wet.
Forward fire room	49.75	55.57
After fire room	49.36	55.51
Forward engine room	29.37	32.58
After engine room	31.76	35.51
Shafting and propellers	15.48	15.48
Tools, stores and spare parts	1.10	1.10
Distilling and heating plants	3.75	3.75
Total weight	180.57	199.50
Detail Weights for One Engine.—		
Cylinders and fittings		6.05
Shafting with propellers		7.74
Reciprocating parts		2.26
Framing, bed plate and bearings		5.63
Valves and valve gear		1.53
Condenser		3.51
Air and circulating pumps		1.05
Boiler and fittings (one boiler without lagging)		17.42
Lagging for one boiler (sheet steel and asbestos)		1.93
Funnel and uptake (one)		1.51
For One Vessel.—		
Steam, exhaust, suction, discharge pipes, and valves		18.37
Lagging and clothing, exclusive of boilers		5.04
Floor plates and gratings		3.62
Auxiliaries		5.97
Fittings and gear		5.48

OBSERVATIONS OF STRAIN DUE TO TEMPERATURE CHANGES AND TO LAUNCHING.

Vessels of the torpedo-boat class being comparatively shallow in proportion to their length and beam, and also very lightly built, must be specially liable to strain. The change at launching from the practically rigid support of the blocks or ways to that of the water might, in some vessels, produce sensible flexure. Also the expansion of the decks and upper sides from sun heat, while the immersed skin is kept very near the temperature of the water, must always produce a constantly varying and complex state of strain.

This effect of changing temperature gives rise to the well known difficulty of lining off the shafting of even large, deep ships, and leads to the frequent and commendable practice of doing this work in the very early morning, before sunrise, at which time the temperature of the mass of the ship is best equalized.

Objects of Observations.—It seemed well to the writers to gain definite knowledge of the variations of strain for two reasons:

1st. To know if the shaft bearings should be relined after launching.

2d. To know to what extent the shafting might be put out of line by diurnal variations of temperature.

No doubt such measurements have frequently been made before, but they have not come under our notice; and we venture to publish the present observations in the hope they will prove interesting and useful.

Changes of strain must be measured relatively to some arbitrarily selected condition of the ship. There is no condition possible which would be free from stress unless we could annul gravity and stresses induced by construction. The observations selected for reference in the *Hopkins* and *Hull* were those in which the conditions gave most equal temperature. In the *Hopkins* these conditions were not very well fulfilled in any observation, and the curves seem to indicate considerable change of strain during the time of this reference observation on the morning of April 23, 1902. In the *Hull* we were more fortunate, and

on May 27th got an observation during very dull weather, in which the air temperature was almost stationary.

Method of Observation.—The method of taking observations and the manner of reduction are extremely simple.

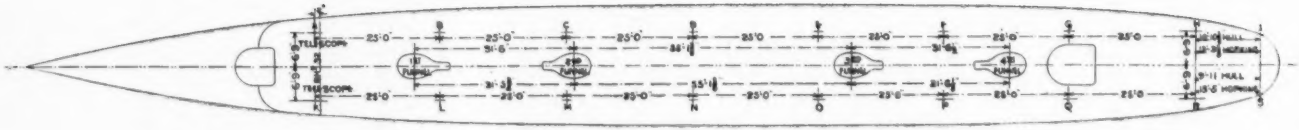
In Fig. 5, A to S, mark stations center punched on the deck, A and K being 12 inches forward of frame 32. Similarly there were center-punch marks at about 5 feet above the deck on each side of each funnel.

An excellent theodolite, by Gurley, of about 7½ inches focal length, was placed at A and K, the plumb bob hanging over the center punch mark and the table of the instrument being leveled. This only approximately brings the telescope to the same position when the ship is on the ways, and afloat with possibly slight variations of trim and heel; but a little consideration will show that the accuracy of the measurements depended to a very trifling extent on even large changes of the telescope station or of the trim and heel of the ship.

The telescope being adjusted at A, two observing staffs were held vertically at B and H, respectively, and the telescope adjusted to give very nearly equal readings at these stations. It was then clamped. During the readings at stations B to I the telescope was not altered in altitude, though to bring the cross-wires on to the staff at the various stations it had to be slightly moved in azimuth by the tangent screw. Measurements were then recorded at each of these stations; all being read from one staff. They were noted to one-thousandth of a foot, and can certainly be relied on within two or three thousandths, except, perhaps, for occasional errors of observation.

The temperatures were taken with thermometers which were carefully compared with a standard certificated thermometer, and found very correct.

Reduction of Observations.—1st. The slight difference of readings at B and H was found, and a proportional correction made in all the readings. That is, if H—B was one-tenth of a foot, one-sixtieth of a foot was subtracted from C, one-thirtieth from D, and so on; B C and B D being respectively one-sixth and one-third of B H.


TELESCOPE
TELESCOPE
1ST FUNNEL
2ND FUNNEL
3RD FUNNEL
4TH FUNNEL
25'-0
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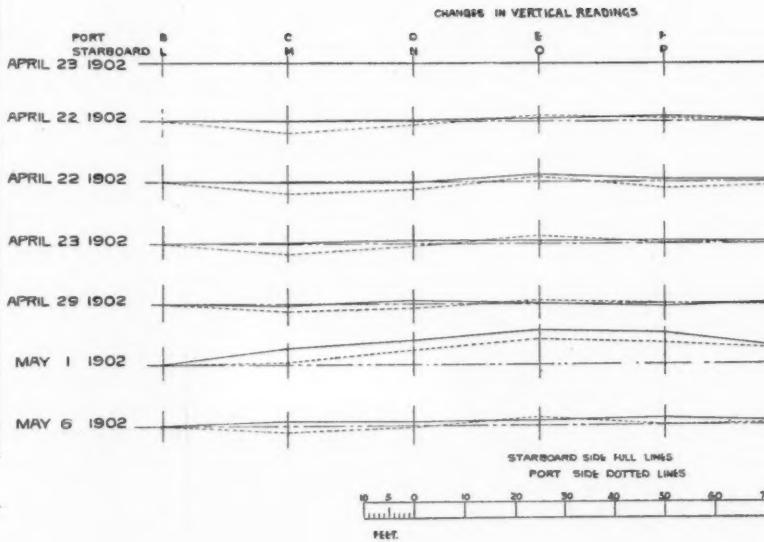
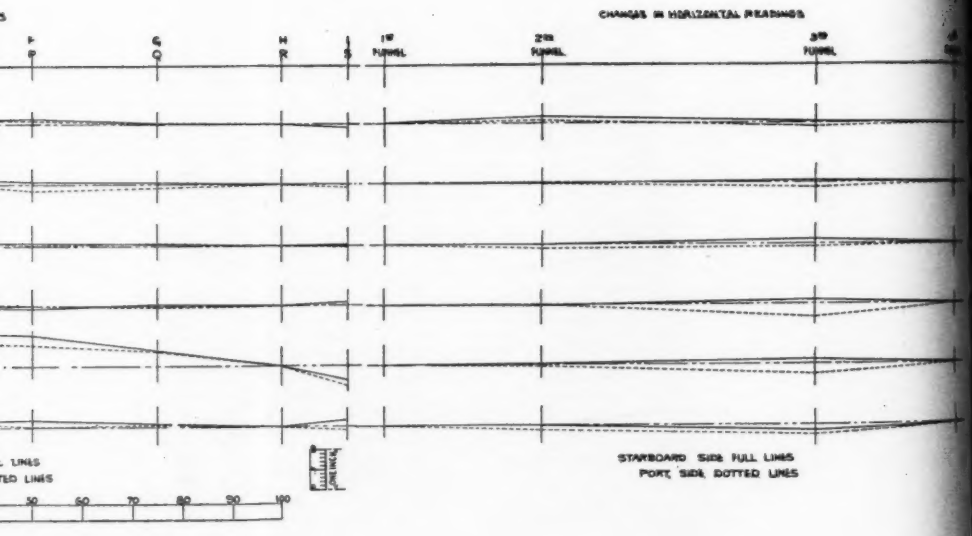
OPKINS."

observation.	Water.	Excess of mean sun over water temperature.	Remarks.
es.	*degrees.*	*degrees.*	Reference observation.
	...	...	Shade, 67 degrees F.....
	...	...	Shade, 79 degrees F.....
	...	...	Shade, 75 degrees F.....
	...	...	.....
7	60	—1	No sun.....
3	65	14	.....
4	65	—3	.....



STRAIN



STRAINS OF "HOPKINS"

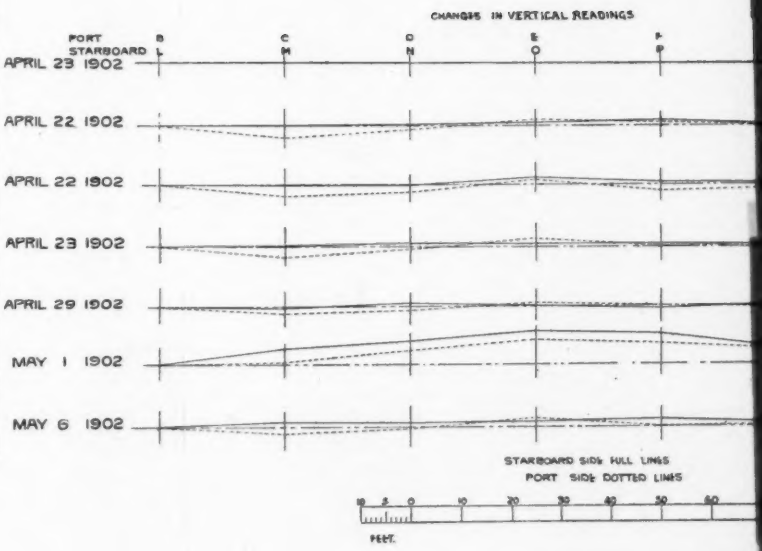
FIG. 6.

STRAIN DEFLECTIONS TAKEN ON U. S. S. "HOPKINS."

Location of boat.	Direction of bow.	Date, 1902.	Observation.		Middle of observation.	Temperatures.						R
						In sun.						
			Commenced.	Ended.		Beginning.	End.	Mean.	Excess over reference observation.	Water.	Excess of mean sun over water temperature.	
						degrees.	*degrees.*	*degrees.*	*degrees.*	*degrees.*	*degrees.*	
On ways.	N. 29° 15′ E.	April 23	6·33 A. M.	7·12 A. M.	6·52 A. M.	71	80	76	0	...	...	Referenc Shade, 6
		April 22	12 M.	12·45 P. M.	12·23 P. M.	84	90	87	11	...	...	Shade, 7
		April 22	5·45 P. M.	6·32 P. M.	6·08 P. M.	84	75	80	4	...	...	Shade, 7
		April 23	1·45 P. M.	2·50 P. M.	2·18 P. M.	94	95	94	18	...	...	.....
In water.	N. 21.5° E.	April 29	12·08 P. M.	12·43 P. M.	12·25 P. M.	59	59	59	—17	60	—1	No sun..
		May 1	12·05 P. M.	*	Say 12·30	79	79	79	3	65	14	.....
		May 6	6·37 A. M.	*	Say 7·05	62	62	62	—14	65	—3	.....

* Time not taken.

sun over water temperature.	Remarks.
...rees.	Reference observation.
...	Shade, 67 degrees F.....
...	Shade, 79 degrees F.....
...	Shade, 75 degrees F.....
...	.....
-1	No sun.....
14	.....
-3	.....



STRAIN

2d. A constant quantity was subtracted from all the readings, which further reduced those at B and H to one foot.

3d. The reference observation having been selected, the change of the reduced reading at C, for any observation, from the reduced reading for the reference observation, gave the strain at C; and similarly for all the other stations, D, E, etc., in this line. Then a line of vertical strain for the starboard side of the ship could be laid down, as in Fig. 6. Similarly the port side and the horizontal measurements from the funnels were treated.

The stations A, B, C, etc., for the *Hull* were in the same positions fore and aft as those for the *Hopkins*, except that H I = 10 feet 10⅜ inches, and R S = 9 feet 11 inches.

The horizontal measurements were not made from the funnels of the *Hull*, but by means of a finely divided scale held at the stations on deck. This seemed much more satisfactory as it was not subject to influence by possible tightening of funnel guys or other forces acting on the thin metal of the uptakes. A change of heel would, however, now have more influence on the observation, but to a very minute degree; and there was no change of heel which would bring this source of error to be of the slightest importance.

Results of Observations.—The *Hopkins* and *Hull* lay side by side on the ways, with their bows pointing 29¼ degrees East of North. The *Hopkins* had the easterly berth. Thus, the morning sun shone on her starboard side, and this was not shaded.

The morning observation of April 23d, 1902, was made very early, with the hope that it would make a good reference observation. But before everything was in order, the sun was already up, and before the conclusion of the observation it was getting quite warm. There is clear evidence in the funnel readings that the heat of the sun, shining full on the starboard side, by expanding the plating, had bent the center part of the ship to starboard. For instance, the observation of May 6th, after the ship was launched, and when she was lying with her bow pointing 21½ degrees east of north, should evidently, from the temperature conditions, have given much less strain than the reference observation; but, the early morning sun being on the starboard

side, it would be bent a little to starboard. The interpretation of the funnel readings is that it was bent less to starboard than during the reference observation of April 23d. The other funnel observations bear the same inference of change during the reference observation.

As each observation occupied a considerable time, change of strain must have been present in nearly all. This accounts for some complexity of the curves from the two sides of the ship, and is most clearly shown in the horizontal observations from the *Hopkins*' funnels. At first glance it would appear that the breadths of the 2d and 3d funnels have increased over the values for the reference observation. But during the reference observation the flexure to starboard was increasing. It was possibly increasing also, but at a much smaller rate, from the weaker sun heat on May 6th; consequently the latter observation shows different lines for the port and starboard sides. If the observations of April 23d and May 6th were taken in the same order, say starboard side then port, the distance between the port and starboard lines from the funnels will be due to the *difference* of the changes during observation on these dates. If the two sets of observations were taken in the reverse order, the divergence of the lines would be due to the *sum* of the changes. The order of the observations was not recorded.

The same peculiarity is shown in the curves of horizontal strain for the *Hull*. Sometimes the lines on the two sides of the deck seem to have approached and sometimes to have receded, and to a much greater degree in many cases than can be assigned to expansion and contraction. They are amply explained by the changing state of strain of the ship and the varied order of making the observations.

It is difficult to assign any satisfactory reason for the large separation of the lines at the third funnel on April 29th and May 1st. It may have been due to errors of observation, but this seems very doubtful as the same peculiarity, though to a less degree, marks the afternoon observations of April 22d and 23d. It seems more likely to be some indirect effect on the light plating of the uptakes and funnels. It does not affect the whole

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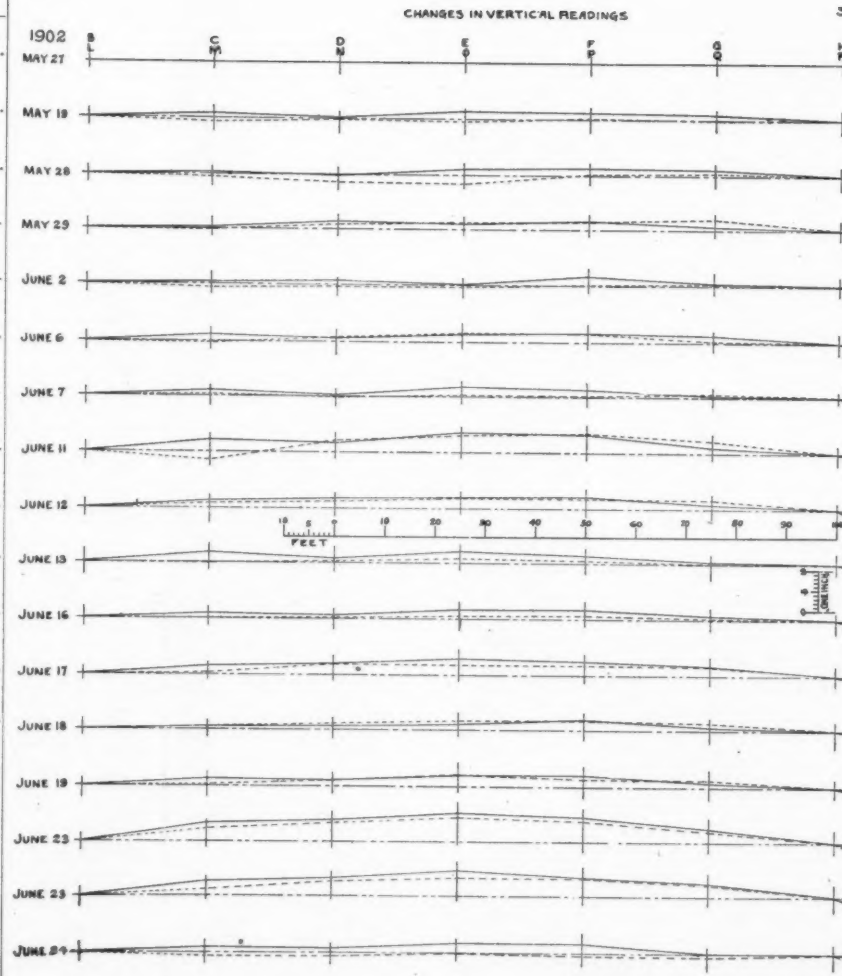
Remarks.
ain from evening before until noon. observation.
y; slight showers from 5·30 to 6·10 P.M.
n. Shade, 64 degrees F.
lear. Water in boilers to 2 inches in
lear. Shade, 74 degrees F.
light showers, 9·55 to 10·05 A. M.
un at intervals only. Ship moved is morning.
lear. Shade, 84 degrees F.
lear. Shade, 86 degrees F. nge of temperature during observation.
lear.
lear. Rain from 9 to 11·30 A. M.
lear. Shade, 81 degrees F.
n 8·45 till 11·25 A. M.
lear. Boat launched June 21.
wer, 3·25 to 4·10 P. M.
lear.



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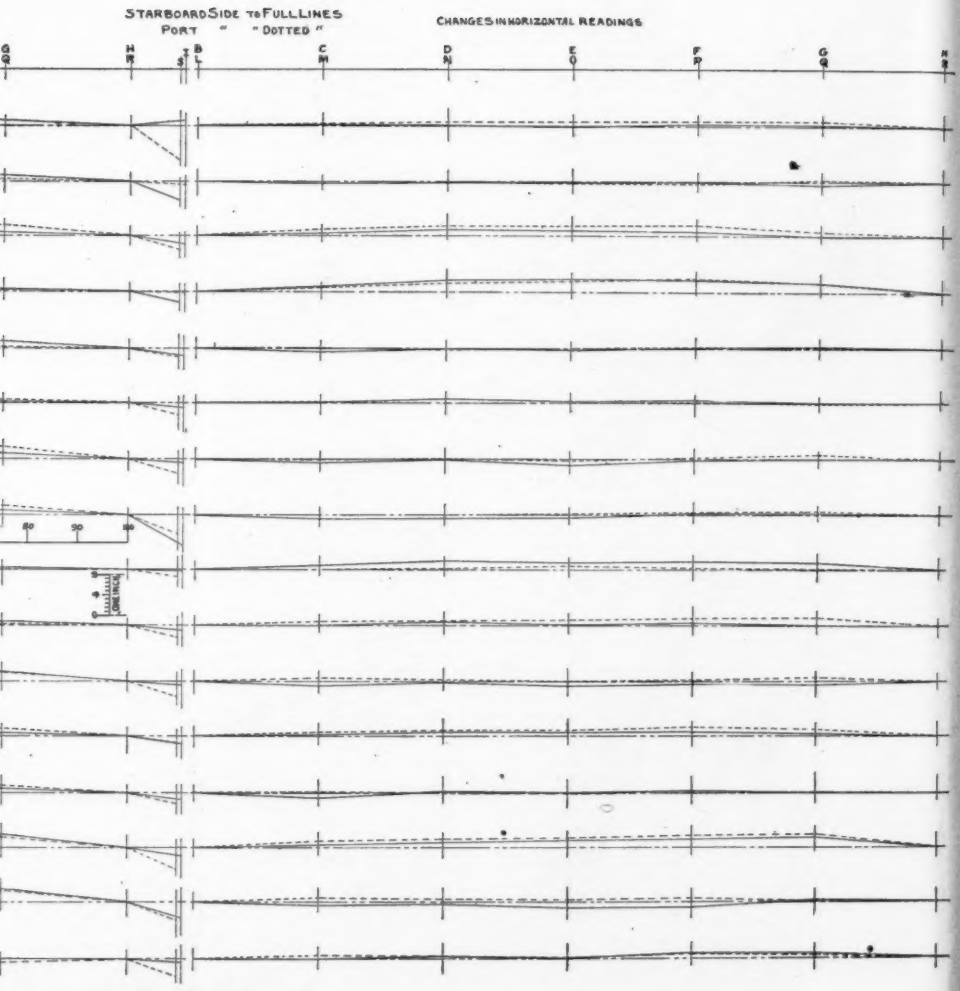


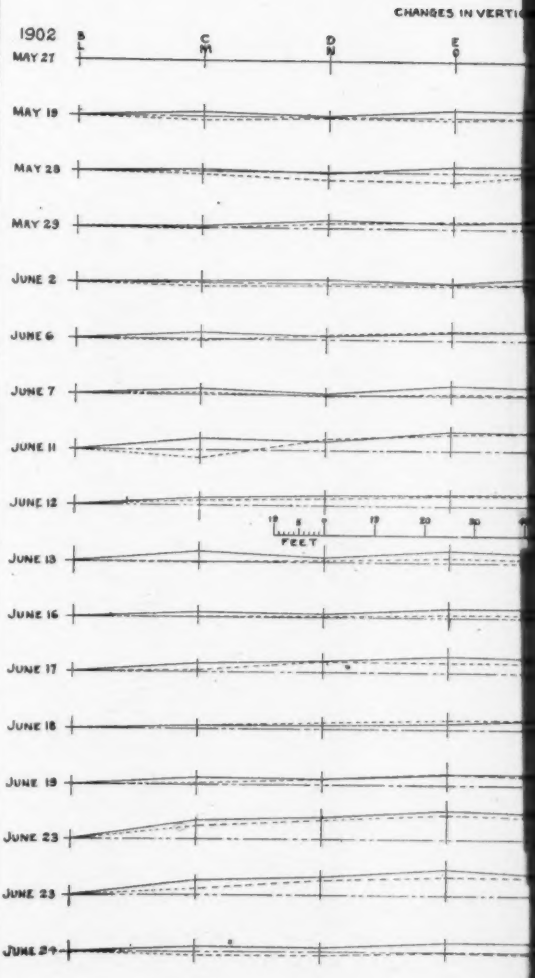
STRAINS OF "HULL"

FIG. 7.

STRAIN DEFLECTIONS TAKEN ON U. S. S. "HULL"

Location of boat.	Direction of bow.	Date, 1902.	Observation.		Middle of observation.	Temperatures.						
						In sun.						
			Commenced.	Ended.		Beginning.	End.	Mean.	Excess over reference observation.	Water.	Excess of mean sun over water temperature.	
						degrees.	*degrees.*	*degrees.*	*degrees.*	*degrees.*	*degrees.*	
Building ways.	N. 29° 15′ E.	May 27	4·15 P. M.	7·10 P. M.	5·42 P. M.	62	58	60	0	...	...	No sun Refere
		May 19	11·50 A. M.	12·42 P. M.	12·16 P. M.	69	69	69	9	...	...	No sun
		May 28	3·30 P. M.	6·45 P. M.	5·07 P. M.	64	56	60	0	...	...	Cloudy
		May 29	9·45 A. M.	1·10 P. M.	11·27 A. M.	79	70	74	14	...	...	Strong
		June 2	9·03 A. M.	11·10 A. M.	10·06 A. M.	90	94	92	32	...	...	Weathe gaug
		June 6	2·32 P. M.	4·33 P. M.	3·32 P. M.	95	93	94	34	...	...	Weathe
		June 7	9·15 A. M.	11·09 A. M.	10·12 A. M.	83	82	82	22	...	...	No sun
Launching ways.		June 11	1·42 P. M.	3·46 P. M.	2·44 P. M.	82	78	80	20	...	...	Cloudy early
		June 12	1·20 P. M.	3·16 P. M.	2·18 P. M.	90	87	89	29	...	...	Weathe
		June 13	6·25 A. M.	8·18 A. M.	7·21 A. M.	73	94	83	23	...	...	Weathe Great c
		June 16	9·45 A. M.	11·41 A. M.	10·43 A. M.	88	89	89	29	...	...	Weathe
		June 17	1·55 P. M.	3·49 P. M.	2·52 P. M.	86	83	84	24	...	...	Weathe
		June 18	9·48 A. M.	11·07 A. M.	10·27 A. M.	79	84	82	22	...	...	Weathe
		June 19	1·28 P. M.	3·27 P. M.	2·27 P. M.	82	84	83	23	...	...	Rain fr
In water.	E. 43° N.	June 23	9·30 A. M.	11·26 A. M.	10·28 A. M.	76	80	78	18	68	10	Weathe
		June 23	2·25 P. M.	4·11 P. M.	3·18 P. M.	76	71	73	13	70	3	Slight s
		June 24	6·30 A. M.	8·09 A. M.	7·20 A. M.	69	80	75	15	67	8	Weathe

Excess of mean sun over water temperature.	Remarks.
degrees.	
...	No sun ; rain from evening before until noon. Reference observation.
...	No sun.
...	Cloudy day; slight showers from 5·30 to 6·10 P.M.
...	Strong sun. Shade, 64 degrees F.
...	Weather clear. Water in boilers to 2 inches in gauge
...	Weather clear. Shade, 74 degrees F.
...	No sun ; slight showers, 9·55 to 10·05 A. M.
...	Cloudy ; sun at intervals only. Ship moved early this morning
...	Weather clear. Shade, 84 degrees F.
...	Weather clear. Shade, 86 degrees F. Great change of temperature during observation.
...	Weather clear.
...	Weather clear. Rain from 9 to 11·30 A. M.
...	Weather clear. Shade, 81 degrees F.
...	Rain from 8·45 till 11·25 A. M.
10	Weather clear. Boat launched June 21.
3	Slight shower, 3·25 to 4·10 P. M.
8	Weather clear.



ship, as a similar result is absent in the *Hull's* curves of horizontal strain, Fig. 7, which were observed at points on the deck.

It may be noted that to diminish the chance of errors of observation, two observers were, during one whole observation, stationed at the telescope to verify each other's readings; but this extended the time of a complete observation so much as to quite change the conditions at the end from those at the beginning. A careful examination of the curves seems to justify confidence in their accuracy.

Having now noted what may be called the mere peculiarities of the curves, we are in a position to examine their more important teaching.

The reference observation of the *Hull*, in the afternoon of May 27, 1902, was made during dull weather, following much rain, and at almost stationary temperature. The conditions could not have been more satisfactory. Hence the teaching of the *Hull's* curves is much clearer than the teaching from the curves of the *Hopkins*. But the results from the two vessels mutually support one another.

The first effect, in a clear morning, is to bend the center of the vessel to the East, which, in these cases, was always the starboard side. This is shown in all the observations on the *Hull*, where the middle of the observation fell before 12 noon. In no other case is there flexure to starboard, except to a comparatively small extent in the observation of May 19th, for which the middle of the observation was 12·16 P. M.

The deflection to starboard is usually accompanied, as would be expected, by the raising of the starboard line on deck. The *Hull* observations of May 29th and June 18th are exceptions for which it is not easy to assign a definite cause. But it is worth remarking that while taking the observations it was strongly realized how impossible it would be to note more than very imperfectly the complex conditions producing strain. Not only would the weather conditions for many hours previous have to be completely described, but also the active operations going forward on board. Even if the conditions could be sufficiently

noted, no one could minutely trace the resulting effects. Unexplainable vagaries of the curves are, then, to be anticipated.

The curves definitely show that, relatively to the vertical flexure, the horizontal flexure may be treated as unimportant. This is due to two things: 1st. The vessel is nearly twice as broad as she is deep, and consequently for a given excess of expansion on one side over the other there will only be about half the curvature that there will be for the same difference of expansion between the deck and the bottom of the ship. 2d. With, say, the ship lying nearly North and South, the East side is exposed to the full force of the sun only in the early morning, and its rays quickly become more oblique, this side passing into shadow about noon. The deck, on the other hand, is exposed to sun heat all day, the rays being nearly direct for a long period on each side of noon, when the sun is most powerful. In fact, in bright, warm weather the bare steel decks of these vessels become intolerably warm during the afternoon. Besides, after the vessel is launched only the top sides are exposed to sun heat, the bottom sides being held from expansion by the water. While this must lead to some warping of the ship, it will usually diminish the flexure. The total curvature of the vessel is compounded of the vertical and horizontal elements, but as these usually reach their maxima at such different times we may safely dismiss the smaller horizontal element.

After the *Hopkins* was launched, ways were laid athwartship under the *Hull*, and early on the morning of June 11th she was drawn across to the launching berth previously occupied by the *Hopkins*. Comparing the afternoon observations of June 6th, 11th, 12th, 17th and 19th, there is decided evidence that the vessel was bent slightly upward in the center by this process. This is most strongly marked on the day the ship was moved, as the upward flexure is greater then in proportion to the difference of temperature than it is in any of the subsequent observations; the inference being that after the vessel was shifted, the center ways, being submitted to the greatest pressure, subsided most. But still the vertical flexures on June 12th, 17th and 19th

are decidedly greater than that of June 6th, though the elevation of temperature, above that of the reference observation, is less.

We will now examine the effect of launching.

For the *Hopkins*, we can best judge from the observation of April 29th. The temperatures of the air and water were sensibly equal and the curves are not bent upward; which means that the vertical flexure of the ship was the same as for the reference observation. Comparing the conditions of the reference observation of this vessel with the morning observations of the *Hull*, particularly with that of June 13th, there probably was some slight vertical flexure in the *Hopkins* on the morning of April 23d. But the inference seems quite sure, that any vertical flexure from launching was very small. The same is taught by the observation of May 6th; for, though the sun was not strong, it would no doubt heat the deck considerably, and vertical flexure would have commenced.

The *Hull* also was not sensibly strained due to changed conditions at launching. Here the reference observation may be considered to have no vertical flexure, and all the flexure shown by the vertical curves on the morning of June 24th is amply accounted for by the conditions on that day.

It only remains to deal with the curves showing very decided vertical flexure.

It might be supposed that when the ship is on the ways it would be prevented from any flexure by the nearly rigid blocks on which it is sitting. But this is far from being the case. Both the *Hopkins* and *Hull* show vertical flexure when the temperature rises. See, for example, the observation of June 6th on the latter, where the elevation of temperature is 34 degrees Fahrenheit. We do not refer to the larger flexure after the shifting of the ship, for, as we have seen, this was partly due to that operation.

This flexure while on the ways indicates both a considerable stress in the structure and a very unequal pressure coming on the blocks. How intense these stresses are would be difficult to calculate with any certainty, as the curves show that the ship is considerably distorted, and no doubt the sections change shape.

When the vessel is afloat and no longer subject to a comparatively inelastic restraint, the flexures for a given elevation of temperature become much greater, as we would expect. This response is also increased by the fact that the bottom is now perfectly protected on the outside from participating in the elevation of temperature. On April 22d, both in the morning and afternoon, the vertical flexure of the *Hopkins* was slight compared with that of May 1st, though the sun heat was greater on the earlier date. And the same is true of the *Hull*, if we compare the observation of June 6th with both those taken on June 23d.

The thermometer cannot be taken as a definite guide in predicting the flexure which the ship is likely to have. The sun heat which the thermometer will show is the temperature at which its absorption balances the losses by radiation and convection; and this depends not only on the state of the glass surface, but on the temperature of the air and the location of the instrument. The same is true of the steel deck, but the latter is more highly absorbent and has much less chance to radiate. On May 1st, while the sun thermometer indicated an excess of 14 degrees Fahrenheit over the water temperature, the vertical deflection corresponds, as will be shown shortly, to an excess of 41 degrees Fahrenheit in the deck over the bottom of the ship, thus requiring a deck temperature of $65 + 41 = 106$ degrees Fahrenheit against 79 degrees Fahrenheit shown by the sun thermometer. This is in line with every-day experience.

Similarly the deck temperature of the *Hull* on the morning of 23d June must also have been about 106 degrees Fahrenheit, while the sun thermometer showed, during the observation, an average temperature of 78 degrees Fahrenheit, or an excess of 10 degrees over the water temperature.

Also on the afternoon observation of June 23d there is still large vertical flexure, though the difference of temperature of the sun and water thermometers was only, on the average, $3\frac{1}{2}$ degrees Fahrenheit. Calculation shows that there was an excess of deck over bottom temperature of about 30 degrees Fahrenheit. This

is decisive proof of the relatively poor radiating power of the steel deck plating.

The most uncomfortable manifestation of this high temperature in the parts exposed to the sun is the great internal heat of the boats themselves.

A possibly serious aspect of the question arises when we consider the effect on the shafting and shaft bearings. As will readily be believed, lining off the shafting of such a ship is difficult, and within certain limits it is guess work; the plain fact being that there is no line possible which can properly be called true, within comparatively large limits. If, when the ship bends, the curvature produced in the shaft is uniform, the induced stress in the worst case shown by these curves (that of May 1, 1902, for the *Hopkins*) is far from serious; but this stress may be much raised by the strains being more or less localized. It seems highly improbable, however, that the effect on the bearings can be other than very detrimental, as the pressure is nearly sure to be localized in one portion or another of the bearing surface, thus producing rapid wear. This is a possible explanation of the fact that both the starboard strut bearings of the *Hopkins* wore down much more rapidly than those on the port side, without any apparent cause. This occurred twice. The curves for both boats show that the flat bottoms near the sterns make these parts especially liable to distortion, and the two sides are not usually affected to the same extent.

We regret that the observations could not be continued longer, but the *Hopkins* trials in Chesapeake Bay began on June 28, and a good opportunity of continuing the experiments never occurred. It is probable that in the warm days of summer, greater deflections than those observed would have been found. But the observations recorded distinctly warn us that there might be trouble if one of this class of boat was run into a cool current in very hot weather. It would be most interesting if the officers in charge of these boats in service would note any circumstances confirmatory or otherwise of this warning.

Thanks are due to Messrs. E. Pier and C. L. Bonham for

assistance in making the observations. Indeed, the large proportion of the observations on the *Hull* were made by them.

Calculations.—The following are the calculations on which some of the foregoing statements are based.

Suppose the vessel curved uniformly between B and H to a radius R and the central deflection to be δ. We have

$\delta \times 2R = \left(\frac{BH}{2}\right)^2$. $BH = 150$ feet; hence taking δ and R in inches, we get

$$R = \frac{75^2 \times 144}{2\delta}. \quad \ldots\ldots\ldots \quad (1)$$

We may take 12 feet as the average distance from deck to bottom of ship, at a point quarter the breadth from the center line. Then if e is the expansion per unit length due to heat, which gives the curvature R, we obviously have

$$\frac{e}{12 \times 12} = \frac{1}{R}. \quad \ldots\ldots\ldots \quad (2)$$

This supposes that originally plane sections remain plane when the ship is bent.

From equation (1), we get

$$e = \frac{\delta}{2{,}812.5}. \quad \ldots\ldots\ldots \quad (3)$$

ϵ, the expansion of steel per 1 degree Fahrenheit, may be taken as $10^{-6} \times 6.35$. Hence the elevation of the deck temperature over that of the bottom for a deflection δ is

$$\frac{e}{\epsilon} = \frac{\delta \times 10^6}{2{,}812.5 \times 6.35} = 56\delta.$$

We may take the temperature of the water as that of the bottom of the vessel also.

Ship.	Date.	δ mean for the two sides.	56δ.	Temp. of bottom and water.	Consequent temp. of deck.	Temp. by thermometer in sun.
				degrees	*degrees.*	*degrees.*
Hopkins....	May 1st,	.74	41	65	$41+65=106$	79
Hull.....	June 23d, A. M.	.68	38	68	106	78
Hull.....	June 23d, P. M.	.54	30	70	100	73½

Suppose the shafting took the same radius of curvature R.

We have $$MR = IE, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where M = the moment of the induced forces in the section, about a transverse line through the center of the shaft.

I = the moment of the section about the same line.

E = Young's modulus of elasticity = say 30×10^6 pounds per square inch, as nickel-steel differs little from mild steel in this constant.

Also $$M = \frac{2fI}{d}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where f = the maximum longitudinal tension and compression in the outer fibers, in pounds per square inch.

d = the shaft diameter in inches.

From equations (4) and (5) we get

$$M = \frac{2fI}{d} = \frac{IE}{R}.$$

Hence

$$f = \frac{dE}{2R} = \frac{\delta dE}{75^2 \times 144},$$

by equation (1).

Taking d as $7\frac{1}{2}$ inches, we have

$$f = \delta \times \frac{7.5 \times 30 \times 10^6}{75^2 \times 144} = 278\delta.$$

For $\delta = .74$, $f = 206$ pounds per square inch. This, combined with the great shearing stress in the material, would raise the total stress very slightly.

TRIALS.

Trial of U. S. T.-B. D. Hopkins in 1902.—The *Hopkins* was launched on April 24, and the *Hull* on June 21, 1902.

On 9th June the *Hopkins* was taken for several hours easy steaming on the Delaware River, but as the water is very shallow no attempt was made to run beyond quite a slow speed. The emergency stop valves and anchor gear were also tried and found satisfactory.

A similar run was made next day.

Everything so far appearing in good order, the *Hopkins* left for Chesapeake Bay on the 14th June.

The first trial on the Barren Island course, Chesapeake Bay, occurred on 28th June.

After two runs on the measured mile the trial was stopped, as the forward H.P. valve gear threatened to heat, and on examination it was found that the steel link block was rubbing against the head of the valve spindle. As it was feared they might fire on and break, it was deemed safer to return to Pearson.

The second trial on Barren Island course was on 3d July, there being six runs over the measured mile.

The highest observed speed was 23.5 knots, giving a true speed of about 23.1 knots, for which the pressures on the engine-room gauges were:

Forward engine, .	224 pounds per square inch.
After engine, . .	217 pounds per square inch.

The revolutions in the quickest run were:

Forward engine, ...	347 per minute.
After engine, ...	299 per minute.
Difference, ...	48 per minute.

There was very considerable vibration of 1st, 2d and 4th period.

As the great difference in the speeds of revolution of the engines was the outstanding feature of the trial, it was determined to go out again and try to force the engines together at full pressure; and the third trial on Barren Island course was run on July 5th.

Only one run was made over the course, and no data was obtained as the vibration became most dangerous. The counters both broke down, but the revolutions of the after engine, by watch, were a little over 300 per minute. This was not very accurately obtained. The speed of the forward engine was very considerably greater, but is not recorded. This trial was made with full boiler pressure—the safety valves blowing and throttles wide open.

It was not thought judicious to continue the trial, as the conditions were so bad and various small fittings, oil cups, etc., had shaken loose. It was decided to brace the engine more completely before running again.

Analysis of the second trial showed that the propulsive efficiency of the after engine was extremely low; for, while it was developing considerable horsepower, the slip of the propeller was much lower than that of the forward engine, which showed that only a small proportion of the energy of the after engine was spent in driving the ship. A simple calculation indicated that somewhere about 2,000 horsepower was being wasted at this propeller. At first it was believed that this waste could only occur from one or more of the propeller blades having been bent, but examination quickly dispelled this belief. It was then thought that possibly it was an indirect effect of torsional vibration. Calculation showed that the natural period for torsional vibration of the after engine was about 618 per minute. Hence this vibration should be worst at a little over 300 revolutions per minute if the vibrations were of second period—that is, if they occurred twice per revolution. With the arrangement of cranks and cylinders adopted in these engines, it is nearly impossible to avoid a second-period impulse in the steam turning moment. There was also a second-period variation in the inertia turning moments. Thus there seemed a strong probability of the pres-

ence of torsional vibration, though even in such severe cases it could not be certainly detected by the eye.

The second-period inertia turning moment could easily be annulled by making the reciprocating masses of the H.P. and I.P. engines equal to those of the two L.P. engines combined; but it was only possible, by shortening the L.P. cuts off, to mitigate this variable impulse from steam. If, however, the second-period element of the turning moment is weakened sufficiently to allow the engines to run together, the damping power of the propeller on these vibrations increases and the vibrations very greatly subside. It was hoped that sufficient improvement could be made to attain this end and, happily, the hope was fully realized.

That torsional vibration is still present to some extent can readily be inferred from Tables II, IV, VII, and IX; for near its critical speed, 300 revolutions or a little over, the after engine runs a few revolutions slower than the forward one, while at higher speeds this characteristic vanishes.

It would be impossible and entirely out of place in this article to enter on the question of torsional vibration, but we may refer the reader to the paper by Herman Frahm, on "Recent Investigations Concerning Dynamical Action in Shafting of Marine Engines, with Special Reference to Synchronous Torsional Vibrations," the translation of which will be found in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XIV, page 721, and we also refer to a very complete discussion in Section V and Appendix II of the articles recently communicated to "Engineering" by our distinguished Engineer-in-Chief.*

We append a long quotation from Section V of this paper, bearing principally on the case of the *Hopkins* and *Hull*, and covering more in detail the same ground as the above brief general statement. This statement will, however, make intelligible some references to an earlier part of Section V.

Trials of U. S. T.-B. D. Hull.—The next trials were those of the *Hull*. After two slow-speed runs on the Delaware River, on

* The "Vibration of Steam Ships," by Rear-Admiral George W. Melville, Engineer-in-Chief of the United States Navy. "Engineering." Vol. LXXV, p. 1.

15th and 19th September, the boat left for Chesapeake Bay on the 27th.

The first trial on Barren Island course occurred on 2d October, when there were eight runs over the measured mile. Just before the last run the air pump of the after engine broke down.

The after engine still ran very well. It may be stated here that on all the trials the forward engine was run from the two forward boilers and the after engine from the two after boilers. Both steam and feed connections were closed at the bulkhead between the engine rooms. This made the work in the fire rooms independent and the regulation of the feed water much simpler. But the condensers were connected through the auxiliary exhaust, which went far to minimize any trouble through derangement of one of the air pumps. No doubt in this case the air escaped from the after to the forward condenser, and the water either accumulated in the condenser or was forced, by slight pressure, through the air pump valves.

Allowing for the tide as deduced from the fifth and sixth runs, the seventh run, with both air pumps, gave 26.38 knots. On the eighth run, with only one air pump, the speed with tidal correction was 25.44. So that the loss of one air pump caused a reduction of speed of less than one knot.

The revolutions per minute were—

	Seventh mile.	Eighth mile.
Forward engine, ...	386	380
After engine, ...	391	362

The air pump, up to this time, was driven by means of a connecting rod attached to the bottom of the trunk, and at top stroke the guidance of the bucket was considerably diminished. When the vessel was turning the slight heel caused the water in the pump to flow to one side, giving a one-sided blow on the bucket at the ends of the stroke. This, it was feared, somewhat upset the bucket at top stroke and caused it to stick. The appearances indicated that the connecting rod drew through the lower bearing and nut, and that on the downward stroke the bucket was punched down. The rim of the bucket had two

cracks, such as seemed to indicate a hard push down; and the bottom of the bucket was also punched down, apparently by scraps of the broken parts getting below the connecting rod.

At this time the air pump design was changed to a yoke and sliding block with ample guides cast on the air-pump covers. Unless the great alteration of fitting independent pumps was to be adopted, this construction seemed the best alternative open. It was adopted, experimentally, and with some misgiving; the parts were made very stiff and the surfaces large. Happily, the result fully justified the experiment.

The second trial on Barren Island course was run on 23d October, there being eight runs over the measured mile. The highest mean speed obtained was 26.38 knots for 8,424 I.H.P. The safety valves were blowing, and the highest revolutions by counter were 396 per minute. The propellers, Figs. 1 and 2, gave 20.1 per cent. mean slip for a pitch of 8.5 feet, and there was evidently much fall of propeller efficiency due to cavitation, as this percentage rose very rapidly at the highest speeds.

Before next trial the Bureau of Construction and Repair was asked to allow the bilge keels to be cut off, for as the ship trimmed these were evidently in a position which would produce much resistance. This having been done, the third trial on Barren Island course took place on 10th November. Seven runs were made. Tables I and II and Figs. 8 to 11 give the principal data for this trial, the fifth run being omitted, as it was spoiled.

TABLE I.—UNITED STATES TORPEDO-BOAT DESTROYER *HULL*. DRAUGHT AND DISPLACEMENT ON TRIAL, NOVEMBER 10, 1902.

	Beginning.	End.
Draught, forward, feet and inches	4–10	4– 7
aft, feet and inches	9– 7½	9– 7
mean, feet and inches	7– 2¾	7– 1
Higher at bow than for even keel, feet and inches	0– 8½*	0–11*
Displacement for half salt water, tons	467†	451†

*These figures mean that the aft reading remaining the same, that at the bow would require to be increased 8½ and 11 inches, respectively, to bring the boat to even keel. It is not for constant displacement.

†Salinometer test of the water on the course showed it to be half salt. It will vary a little with the state of the tide. It is assumed here that the density is the same at Pearson, where the draughts were read.

TABLE II.—UNITED STATES TORPEDO-BOAT DESTROYER *HULL*. SPEED AND POWER DATA OF TRIAL ON NOVEMBER 10TH, 1902.

Run.	Tide.	Revolutions per minute.				Indicated horsepower.				Speed.		Mean slip, 8.5 ft. pitch.		Steam at engines.		Vacuum.	
		Forward engine.	After engine.	Mean.		Forward engine.	After engine.	Total.	Mean.	Single run.	Double run.	Single run, per cent.	Double run, per cent.	Forward engine.	After engine.	Forward engine.	After engine.
				Single run.	Double run.												
1	Against	307	300	303.5	306	1,801	1,713	3,514	3,640	21.75	22.19	14.6	13.5	125	125	27.2	28
2	With	314	302	308		2,016	1,751	3,767		22.64		12.4		127	125	27.2	28
3	Against	357	344	350.5	357.5	3,290	2,454	5,744	6,061	24.65	25.27	16.2	15.7	200	170	26	27.7
4	With	365	364	364.5		3,193	3,185	6,378		25.89		15.3		200	197	25.5	27
6	With	412	419	415.5	412	4,635	4,484	9,119	8,866	28.01	27.77	19.6	19.6	270	255	23.5	26
7	Against	410	406	408		4,456	4,158	8,614		27.53		19.6		265	250	23.5	26

Forward engine, H.P. cut-off..... .731
After engine, H.P. cut-off..... .724
Mean, H.P. cut-off..... .727

During the seventh run the coal in the after bunkers was exhausted, so that the after engines fell in revolutions.

This trial gives the maximum revolutions that were attained

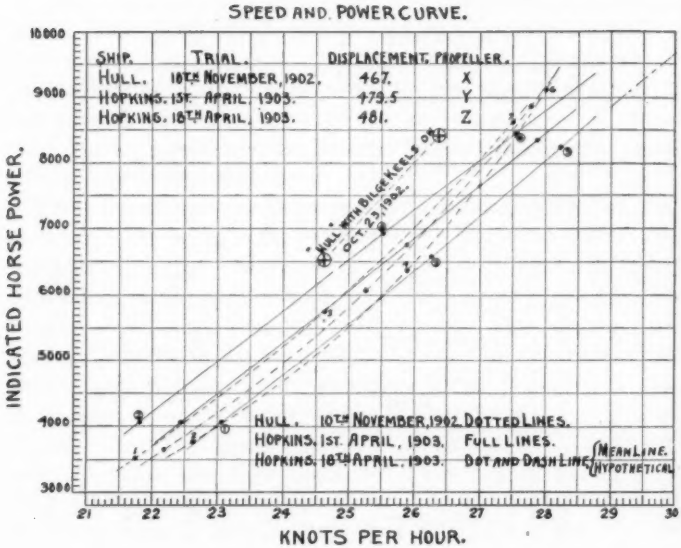


FIG. 8.

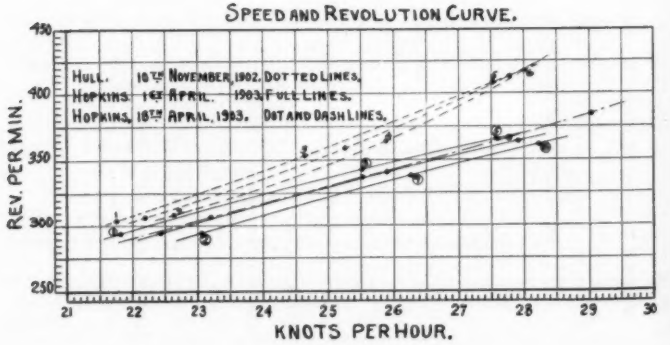


FIG. 9.

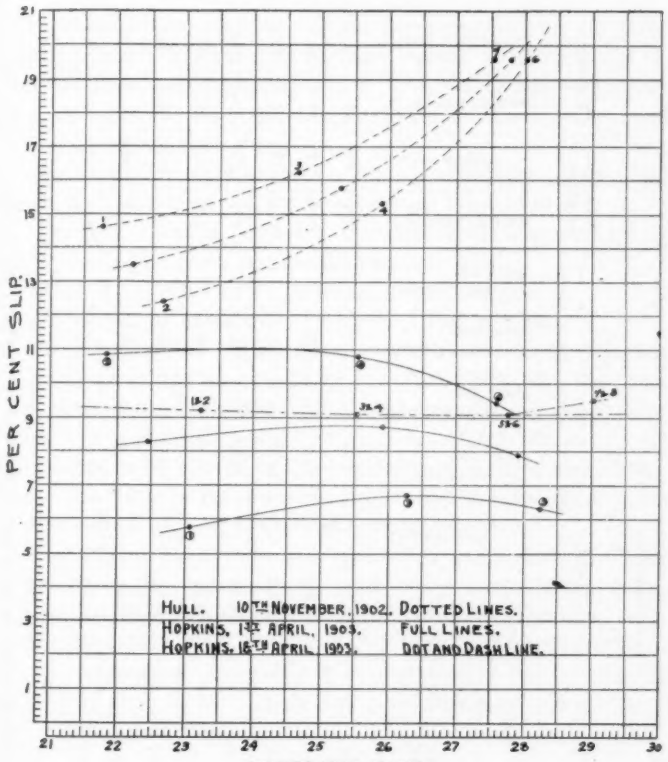

PER CENT SLIP.
KNOTS PER HOUR.
HULL. 10TH NOVEMBER, 1902. DOTTED LINES.
HOPKINS. 1ST APRIL, 1903. FULL LINES.
HOPKINS. 18TH APRIL, 1903. DOT AND DASH LINE.

FIG 10.

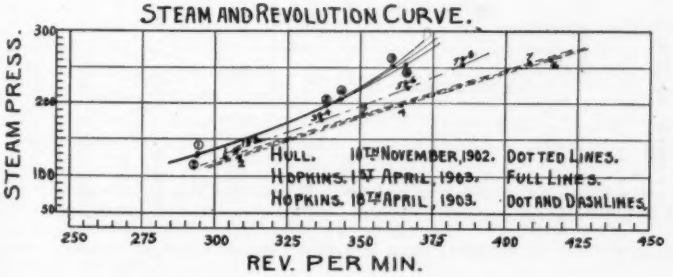

STEAM AND REVOLUTION CURVE.
STEAM PRESS.
REV. PER MIN.
HULL. 10TH NOVEMBER, 1902. DOTTED LINES.
HOPKINS. 1ST APRIL, 1903. FULL LINES.
HOPKINS. 18TH APRIL, 1903. DOT AND DASH LINES.

FIG.11.

TABLE IV.—UNITED STATES TORPEDO-BOAT DESTROYER *HULL*. SPEED AND ENGINE DATA OF STANDARDIZING TRIAL.

Run.	Tide.	Revolutions per minute.				Speed.		Mean slip.*				Steam at engines.		Vacuum.	
								8.27 ft. pitch.		8.5 ft. pitch.					
		Forward engine.	After engine.	Mean.		Single run.	Double run.	Single run, per cent.	Double run, per cent.	Single run, per cent.	Double run, per cent.	Forward engine.	After engine.	Forward engine.	After engine.
				Single run.	Double run.										
1	With	318	300	309	310	22.76	22.41	9.8	11.5	12.2	13.9	127	126	25	21.3
2	Against	315	308	311.5		22.06		13.2		15.6		129	126	25.5	24
3	With	364	359.5	362	361	26.05	25.61	11.8	13	14.2	15.4	183	187	22.8	20.5
4	Against	358	361	359.5		25.17		14.2		16.5		187	184	23.8	22.3
5	With	405	404.5	405	398	27.73	27.18	16.1	16.3	18.4	18.6	233	225	19.7	19
6	Against	380	401	390.5		26.63		16.4		18.7		221	237	5.3	19.7
7	With	410	416	413	412.5	28.21	28.03	16.3	16.7	18.6	19	252	249	.3	18.8
8	Against	406	417.5	412		27.86		17.2		19.4		255	250	1	18.1

Forward engine, H.P. cut-off, .748. After engine, H.P. cut-off, .742. Mean, H.P. cut-off, .745.

* We calculate the slip for two pitches, for reasons given earlier under the headings of "Propellers."

with these boats, 419 per minute corresponding to 1,257 feet piston speed. Even then the boilers seemed by no means at the limit of their steaming capacity, as the pressure was maintained with comparative ease, and the safety valves lifted. The mean H.P. cut-off was then .727.

The propeller again shows a poor performance at the high speeds, as may most readily be seen from the curves, Fig. 8.

The good effect of removing the bilge keels is shown by the change of squat from that for the previous trial:

October 23, squat at 26.38 knots, 5 feet 8 inches.

November 10, squat at 27.77 knots, 5 feet 3 inches.

The squat at the higher speed was thus smaller by 5 inches.

Application was now made for the official trial, and the vessel was docked and repainted with McInnes' paint. A slight alteration was also made in the feed suctions, which allowed all the feed water to be carried in the forward feed tank, in order to improve the trim. Also the main engine valves were faced off to give a cut-off of .745, as there was steam to spare.

The official trials were thus the third and fourth in Chesapeake Bay, the standardizing trial being over Barren Island course, and the endurance trial in the channel of the Bay.

Standardizing Trial of U. S. T.-B. D. Hull.—This trial was run on the morning of December 6, commencing at 10·15 A. M. and ending at 11·25 A. M. There was a light breeze from northeast by east and a smooth sea.

TABLE III.—UNITED STATES TORPEDO-BOAT DESTROYER *HULL*.
DRAUGHT AND DISPLACEMENT ON STANDARDIZING TRIAL.

	Beginning.	End.*
Draught, forward, feet and inches	4-10	4- 2¼
aft, feet and inches	9- 6¼	9- 6¾
mean, feet and inches	7- 2¼	6-10¾
Higher at bow than for even keel, feet and inches	0- 7¼	1- 3¼
Displacement for half salt water, tons	465.5†	432.5

* The forward boilers were blown down before the boat returned to Pearson, and consequently these figures are not true for the end of the runs. These two boilers contain 4.78 tons of water, at 416 degrees Fahrenheit (230 pounds pressure), up to steaming level.

† Our reading of the displacement curve for the figures at the beginning, allowing for change of trim from even keel, gives 459 tons.

The data of this and the endurance trials are compiled from the official report, and are given in Tables III to V, from which parts of Figs. 8 to 11 are constructed.

Compensating weights were put aboard to comply with the contract requirements, given earlier.

On or immediately before the sixth run the vacuum of the forward engine dropped suddenly. It was found on examination, two days after the trial, when there was time to examine the machinery, that the bucket had completely severed from the trunk in the air pump of the forward engine. But both official trials were completed successfully notwithstanding this drawback. As in the trial of 2d October, no doubt the air escaped to the after condenser—indeed this seems to be indicated by the vacuum for this engine—and the air-pump trunk was ample to pump out the water.

The throttles were wide open.

The slip is calculated both for 8.27 feet pitch and 8.5 feet, for reasons given in the paragraph referring to the propeller. There seems no doubt that cavitation had set in before the highest speed runs, and the greater horsepower which would have been obtained had the air pump held intact would probably have added considerably to the slip and little to the speed of the vessel. It may be a matter of surprise that there was not a much greater decrease of revolutions in the after engine, but it must be remembered that the horsepower will be roughly proportional to the square of the slip. The slips of each engine for the sixth, seventh and eighth runs and 8½ feet pitch are—

Run.	Forward engine.	After engine.	Ratios of slips squared.
6	16.4	20.8	$\left(\frac{16.4}{20.8}\right)^2 = .622$
7	18	19.2	$\left(\frac{18.0}{19.2}\right)^2 = .879$
8	18.2	20.4	$\left(\frac{18.2}{20.4}\right)^2 = .796$
Mean of 6, 7 and 8.....			.766
Mean of 7 and 8.....			.838

At these high speeds all cylinders gave nearly equal power; consequently that from the two forward L.P.s was about 25 per cent. of the whole. Cutting out the vacuum in the forward condenser would take away about one-half of the power of these two cylinders, or say 12 per cent. of the whole power of the two engines combined, leaving about 88 per cent. This agrees fairly with the means from runs 7 and 8, given above, when we remember that taking account of the wake would considerably raise the means. On the other hand, had the after vacuum not been affected the means would have been somewhat lowered. The ratio, .622, from run 6, is evidently an exceptionally low result, partly due to the lower steam pressure at that time in the forward engine room.

This is, of course, only put forward as a very rough calculation, as it might be affected by other circumstances besides those noted. It is merely given to show that the comparatively slight falling off in revolutions of the forward relatively to the after engine when the air pump broke, was about what might be expected.

Endurance Trial of U. S. T.-B. D. Hull.—This trial was also made on the 6th December, 1902, commencing at 5'01 P. M. and lasting for one hour. The previous trial had given 372 revolu-

TABLE V.—UNITED STATES TORPEDO-BOAT DESTROYER *HULL*. DATA OF ONE HOUR'S ENDURANCE TRIAL.

Time,	Revolutions per minute.		Steam at engines.		Vacuum.	
P.M.	Forward engine.	Aft engine.	Forward engine.	Aft engine.	Forward engine.	Aft. engine.
5'01	...	...	230	220	0	21
5'11	383.6	384.2	235	230	0	21
5'21	391.1	385.4	240	225	2	21.5
5'31	392	394.1	235	226	2	20.5
5'41	392.3	397	245	226	2	19.5
5'51	400.9	398.4	245	225	1	21.5
6'01	402.5	392.1	255	225	2	22
Mean	393.7	391.9	240.7	225.3	1.3	21
	391.9					
	2) 785.6 (392.8 mean for both engines.					

tions per minute as requisite for 26 knots. There was a light breeze from northeast by east and a smooth sea. The vessel was in precisely the same condition as regards draught readings and displacements as for the morning's trial, Table III. The data of the run are given in Table V.

The mean speed for this trial is 27.07 knots. After these trials the steering gear was tested, both when going ahead and astern. It was found satisfactory and the gear showed no sign of weakness.

The vibration of the vessel is reported to be at all times moderate; at no point on deck or in the engine foundations showing local vibrations which were excessive. The most marked vibration was found over the propellers, "but it was not excessive even at this point."

The vessel returned to the contractors' yard, where she was finished, and delivery at League Island Navy Yard was made on 18th March, 1903.

Fourth Preliminary Trial of U. S. T.-B. D. Hopkins.—The trials of the *Hopkins* were now continued, and her fourth on Barren Island course took place on 1st April, 1903.

A new propeller of 8 feet 6 inches diameter, Figs. 3 and 4, had been fitted, from which much better results were anticipated and realized.

There were six runs over the course. The data are given in Tables VI and VII and Figs. 8 to 11.

TABLE VI.—UNITED STATES TORPEDO-BOAT DESTROYER *HOPKINS*.
DRAUGHT AND DISPLACEMENT ON TRIAL, 1ST APRIL, 1903.

	Beginning.
Draught, forward, feet and inches	5–2¼
aft, feet and inches	9–6¾
mean, feet and inches	7–4½
Higher at bow than for even keel, feet and inches	0–3½
Displacement for half salt water, tons	479.5

The draughts at the end of the trial were not taken.

There is a considerable discrepancy between the displacement of the *Hull* on her trials of 10th November and 6th December on the one hand, and that just given for the *Hopkins*. It has

TABLE VII.—UNITED STATES TORPEDO-BOAT DESTROYER *HOPKINS*. SPEED AND POWER DATA OF TRIAL ON 1ST APRIL, 1903.

Run.	Tide.	Revolutions per minute.				Indicated horsepower.				Speed.		Mean slip.*		Steam at engines.		Vacuum.	
		Forward engine.	After engine.	Mean.		Forward engine.	After engine.	Total.	Mean.	Single run.	Double run.	Single run.	Double run.	Forward engine.	After engine.	Forward engine.	After engine.
				Single run.	Double run.												
1	With	298.5	289	294	294	2,070	1,981	4,051	4,053	23.07	22.44	5.74	8.27	135	125	26	27.2
2	Against	299	288	293.5		2,061	1,995	4,056		21.81		10.81		131	127	26	27.2
3	With	347	329	338	341	3,460	3,116	6,576	6,750	26.27	25.91	6.68	8.73	202	192	26.7	27
4	Against	351	336	343.5		3,533	3,392	6,925		25.53		10.78		213	200	27.5	27
5	With	368	355	361.5	363.5	4,422	3,821	8,243	8,349	28.23	27.91	6.27	7.84	251	250	26	26.7
6	Against	373	358	365.5		4,463	3,993	8,456		27.58		9.42		255	250	26	26.5

Forward engine, H.P. cut-off..... .721
After engine, H.P. cut-off..... .720
Mean, H.P. cut-off..... .720

* The slip of the propeller of the forward engine is calculated for a pitch of 8.38 feet, and that of the after engine for a pitch of 8.51 feet.

been carefully examined, but a clear explanation has never been found. It is not an error of reading, as the differences of the *Hull's* readings agree with the known differences, and it was a matter of careful thought why we could not bring the *Hopkins* before trial to the same readings. Errors of placing the draught figures could not account fully for the fact, nor, should we think, would differences in the lines of the ship. The large proportion of the discrepancy must be accepted as real.

The highest result possible was not attained. Several times on the fifth and sixth runs the safety valves lifted, and to save feed water the boiler-room hatches were opened more than once. This caused considerable irregularity.

But, nevertheless, comparison with Table II shows that a rather higher speed was attained for fully 500 I.H.P. less, and the propeller shows no sign of cavitation. The result, for equally efficient propellers, should have been considerably against the *Hopkins*, as the bottom was in very poor condition, while that of the *Hull* was painted a few days before with McInnis' anti-fouling paint. Besides, between Tables I and VI there is the difference of 12.5 tons in the displacement at the beginning of the trial, also against the *Hopkins*.

One very satisfactory feature was that the result was gained with 48.5 revolutions—145.5 feet piston speed—per minute less than on 10th November. As anticipated in designing the propeller, it held the engine down rather much, and it was decided to cut it down to 8 feet diameter.

As the result was so certain to bring the vessel much above 28 knots, it was not considered necessary to have any more preliminary trials, and the official trial was asked for. These were the fifth and sixth in Chesapeake Bay.

Standardizing Trial of U. S. T.-B. D. Hopkins.—The trial commenced at 12·19 P. M., April 18th, and ended at 1·35 P. M. There was a gentle breeze from the northwest and a smooth sea.

The data of this and the endurance trial are compiled from the official report, and are given in Tables VIII to X, and the results of the standardizing trial are shown in Figs. 8 to 11.

This vessel was weighted to comply with the contract requirements.

TABLE VIII.—UNITED STATES TORPEDO-BOAT DESTROYER *HOPKINS*. DRAUGHT AND DISPLACEMENT ON STANDARDIZING TRIAL.

	Beginning.	End.
Draught, forward, feet and inches	5-2	4-9
aft, feet and inches	9-7½	9-6½
mean, feet and inches	7-4⅝	7-1⅞
Higher at bow than for even keel, feet and inches	0-4½	0-8½
Displacement for half salt water, tons	481	453
Weights from records plus trial weights, tons	480.51	

Endurance Trial of U. S. T.-B. D. Hopkins.—Later in the afternoon of the same day, 18th April, 1903, the *Hopkins* left Pearson for her endurance trial. This commenced at 5·16 P. M. and finished at 6·16 P. M. She was run for 20 minutes southward in Chesapeake Bay, then turned with a small helm and run for the balance of the time northward. The sea was smooth with a light westerly breeze. The vessel was in the same trim and had the same displacement when starting, as is given by Table VIII for the commencement of the standardizing trial. The data of the run are given in Table X.

TABLE X.—UNITED STATES TORPEDO-BOAT DESTROYER *HOPKINS*. DATA OF ONE HOUR'S ENDURANCE TRIAL.

Time.	Revolutions per minute.		Steam at engines.		Vacuum.	
P. M.	Forward engine.	Aft engine.	Forward engine.	Aft engine.	Forward engine.	Aft engine.
5·16	...	...	170	182	27.5	27
5·26	355	340.6	210	215	26	26.5
5·36	374.1	354.1	215	217	26	26.5
5·46	371.7	355.9	205	218	26	26.2
5·56	361.2	349	200	217	25.5	26
6·06	358.2	347	198	205	26.5	25.2
6·16	348.4	326.3	205	170	26	26.5
Mean.	361.43	345.48	200.4	203.4	26.2	26.3
	345.48					
	2)706.91 (353.455 mean for both engines.					

The curve obtained from the standardizing trial having shown that 343 revolutions are requisite for 26 knots, the speed of this trial was given as 26.7 knots per hour.

TABLE IX.—UNITED STATES TORPEDO-BOAT DESTROYER *HOPKINS*. SPEED AND ENGINE DATA OF STANDARDIZING TRIAL.

Run.	Tide.	Revolutions per minute.				Speed.		Mean slip.*		Steam at engines.		Vacuum.		Air pressure.	
		Forward engine.	After engine.	Mean.		Single run.	Double run.	Single run.	Double run.	Forward engine.	After engine.	Forward engine.	After engine.	Forward stokehold.	After stokehold.
				Single run.	Double run.										
														INS.	INS.
1	With	304	301	302.5	307	23.97	23.21	4.9	9.2	129	134	28	27.5	1	3
2	Against	313	309	311		22.44		13.4		139	133	28	27.3	2.25	4
3	With	343	333	338	337	26.67	25.53	5.3	9.1	181	172	27.3	27	6	5
4	Against	342	330	336		24.39		12.9		185	161	27.5	27.3	4	1.5
5	With	368	355	361.5	366.5	28.57	27.76	5.1	9.1	217	212	26.2	26.62	6.75	7.25
6	Against	373	370	371.5		26.95		13		225	224	26	26.5	7	7.25
7	With	385	374	379.5	385	29.7	29.02	6.1	9.5	244	234	25.5	26.3	7.5	8
8	Against	394	387	390.5		28.35		12.9		262	262	25.5	25.5	8.25	8.5

Forward engine, H.P. cut-off, .738. After engine, H.P. cut-off, .748. Mean, H.P. cut-off, .743.

* The slip of the propeller of the forward engine is calculated for a pitch of 8.38 feet and that of the after engine for a pitch of 8.51 feet.

The report regarding slightness of vibration of the vessel is the same as that for the *Hull*. Naval Constructor Groesbeck's report proceeds:

"The engine foundations were carefully examined as far as practicable and the working of the engines observed during the standardizing runs, and I have to report that the absence of vibrations in the same was remarkable. Even at the highest speeds, the vibrations were extremely moderate."

The vessel, after being finished at the contractors' yard, was delivered at League Island Navy Yard on 27th May, 1903.

REMARKS ON TRIALS.

True Speed of Hopkins on Standardizing Trial.—The official trials of the *Hopkins* were unfortunately run with poor coal. This had been bought expressly for these official trials to insure its being fresh and good, but it turned out to be nearly all slack. For this reason the boilers could not be made to give steam freely, and near full power the pressure could only be raised slowly. This is the reason for the low steam pressure of the 7th standardizing run, Table IX. Even on the 8th run full pressure was not obtained and the safety valves showed no sign of lifting—full pressure in the boilers showed 270 pounds in the forward engine-room gauge, Table II.

As a consequence, the 7th and 8th runs were made at very different true speeds, and it is of interest to know what the actual speed of the 8th run was, as obviously it was considerably above the mean, 29.02.

The strength of tide may readily be found very approximately, and as it is considerably above what we are informed is the usual maximum strength for Barren Island course, we will give its value for each pair of runs, to show that the whole data are consistent and that we are not being misled by some undiscovered error of observation.

The slip is nearly constant and, therefore, within very narrow limits of error, we may, when comparing any one pair of runs, consider the true speed of each to be proportional to the number of revolutions per minute.

Taking runs 1 and 2, and putting T for the strength of tide in knots, we have:

True speed of run 1, with tide $= 23.97 - T$; 302.5 revolutions.

True speed of run 2, against tide $= 22.44 + T$; 311 revolutions.

Hence $$\frac{23.97 - T}{22.44 + T} = \frac{302.5}{311}.$$

This gives $T = 1.09$.

Calculating similarly from all the pairs, we get:

Runs 1 and 2, tide in knots, ...	1.09
Runs 3 and 4, tide in knots, ...	1.06
Runs 5 and 6, tide in knots, ...	1.19
Runs 7 and 8, tide in knots, ...	1.09

The slight variations shown are no doubt due partly to the natural change of strength of the tide during the short time of trial and to eddies, and also to effects produced by the engines running together less perfectly at one time than another. Accepting the value 1.09 for the last pair of runs, which certainly does not seem to be an overestimate, the true speed of the 8th run is $28.35 + 1.09 = 29.44$. This agrees with the value of 29.45 arrived at by the trial board and given to us unofficially.

The present writers each took the time independently, and while agreeing very fairly with the Board's time on the earlier runs, they differed by over half a second on the last two runs. Calculating as above from our observation gave the true speed of the 8th run about 29.6. This is supported by the rise of slip on the last runs which appears as a discontinuity in the slip curve. This discontinuity cannot be explained by cavitation, as the pressure per square foot had not risen to the value at which this usually occurs. This we will see immediately. Nor is it to be explained by the inequality of speed on runs 7 and 8. Taking 29.44 as the true speed, we get for the propeller of the forward engine, at 394 revolutions per minute and 8.38 feet pitch, a slip of—

$$100\left(1 - \frac{29.44 \times 6{,}080}{394 \times 8.38 \times 60}\right) = 9.65 \text{ per cent.}$$

Similarly the propeller of the after engine gives 9.44 per cent., from which we get a mean slip for run 8 of 9.54 per cent. Run 7 at 28.61 gives 9.53 per cent. mean slip. But if we rate run 8 at 29.6 and raise run 7 by the same amount, namely to 28.77, we get:

Mean slip for run 8 = 9.05 per cent.
Mean slip for run 7 = 9.03 per cent.

These last values are much more consistent with the rest of the trial data, and leave no doubt that 29.6 was approximately the highest true speed.

This speed was attained notwithstanding the large overweight of 73 tons—just barely 18 per cent.

Mean Pressure referred to Low-pressure Cylinders and Indicated Horsepower per Revolution.—The mean pressure referred to the two L.P. cylinders and the indicated horsepower per revolution—if we may use a possibly illogical, but convenient term—are proportional to one another. The best results attained are given in Table XI.

TABLE XI.—PERFORMANCE OF ENGINES.

Ship.	Table.	Run.	Engine.	I.H.P.	Pressure at engine.	Revolutions per min.	H.P. cut-off.	Mean pressure referred to L.P. cylinders.	Ratio of mean pressure to absolute pressure at engines.	I.H.P. per revolution.
Hull.....	II	6	For'd	4,635	270	412	.731	68.7	.241	11.25
Hull.....	II	6	Both	9,119	262	415.5	.727	67	.242	21.95
Hopkins....	VII	5	For'd	4,422	251	368	.721	73.4	.276	12.02
Hopkins....	VII	6	Both	8,456	252	365.5	.72	70.6	.265	23.14

Performance of Boilers.—The best result attained—except, probably, in run 8 of the standardizing trial of the *Hopkins*,

when no indicator diagrams were taken—was in the *Hull*, on November 10th, run 6, Table II.

$$\text{Forward engine, } \frac{4{,}635 \text{ I.H.P.}}{147 \text{ sq. ft. G.S.}} = 31.5.$$

$$\text{Both engines, } \frac{9{,}119 \text{ I.H P.}}{294 \text{ sq. ft. G.S.}} = 31.$$

As stated earlier, the forward engine was run from the forward boilers alone.

In no trial, except when standardizing the *Hopkins*—the coal then being bad—did the boilers seem very near the limit of their steaming capacity, as the safety valves lifted frequently. Nor did our excellent crew of stokers require to exert themselves to their fullest extent to do this. There was certainly a considerable percentage of power unused. After the trial of 10th November, we lengthened the cut-off of the *Hull's* engines by cutting down the H.P. steam laps as far as we dared, to utilize some of this surplus on official trial. Unfortunately the breakage of the air pump prevented the desired result being realized.

Performance of Propellers.—The indicated thrust per square foot at various times is given in Table XII.

TABLE XII.

Vessel.	Table.	Runs.	Propeller.	Pitch.	Projected area, two propellers.	Revolutions.	I.H.P.	Thrust in pounds.	Thrust in pounds per square foot.
Hull.....	II	3 and 4	X	8.5	35	357.5	6,061	65,820	1,881
Hull.....	II	6 and 7	X	8.5	35	412	8,866	83,540	2,387
Hopkins...	VII	5 and 6	Y	8.44 Mean.	44	363.5	8,349	89,800	2,041
Hopkins...	IX	8	Z	8.44 Mean.	42.5	390.5	Say 9,372	93,840	2,208

It is impossible from Fig. 8 to say where propeller X began to fall seriously in efficiency. But unless the response of the ship to increase of power is very slight for a considerable range near the highest speed, the commencement of failure must have been but little above runs 3 and 4 and at a thrust per square foot not much, if any, above 2,000 pounds, which is unusually early. Our reasons for fearing this result might happen are given earlier, under "Propellers."

If the hypothetical dot-and-dash curve, Fig. 8, for the standardizing trial is near correct, there seems to have been a gain of speed, due to repainting the ship, of between .4 and .5 of a knot, making some allowance for the slightly greater displacement on the official trial. Also, even making no allowance for the 14 tons greater displacement in the *Hopkins*' official trial over that of the *Hull*, on 10th November, 1902, the dot-and-dash line of Fig. 8 would indicate no advantage in propeller X even at low powers.

The notching of the propellers spoken of in the appended extract from Admiral Melville's paper, showed very markedly in the *Hopkins*' propellers after the trials of 3d and 5th July, 1902. After the second-period torsional impulse had been mitigated, it was found that the *Hull* only showed a trace of it on one blade, thus tending to confirm the surmise that the notching was due to severe torsional vibration. When designing propeller Y we had no hesitation in running the sections almost to a knife edge both at the leading and following edges, and we found after trial that no trouble had resulted.

Effect of Cutting Off the Bilge Keels.—The results with the bilge keels is indicated by the two marks ⊕, Fig. 8, from the data of the *Hull's* trial, on 23d October, 1902. The draughts and displacement for the beginning of this trial were—

Draught, forward, feet and inches	4-10¼
aft, feet and inches	9- 8¾
mean, feet and inches	7- 3 7/16
Higher at bow than for even keel, feet and inches	0- 9⅜
Displacement for half salt water, tons	474

There is only 7 tons increase of displacement over that on November 10, 1902; hence the difference of about one knot is due almost entirely to the bilge keels, as the bottom in both cases was in good condition. This shows, as appeared probable from examination of the ship in drydock, that the line of bilge keels was unsuitable for the trim in which the ship had to be run. It does not follow that such a great drag is always caused by bilge keels when on a suitable location; indeed, we are assured that tank experiments show but very slight increase of resistance above that due to the small increase of wetted surface when the keels are properly placed.

What Speed could be Realized with these Boats?—It may not be uninstructive to enter a short distance into the land of speculation to settle what would have been the next step had it been necessary to obtain a higher speed. We have seen from Table II that the boilers gave as high as 31.5 I.H.P. per square foot of grate. It is quite safe to assume that there was a margin of more than 5 per cent.; that is, that $31.5 \times 1.05 = 33.1$ horsepower per square foot of grate could readily be obtained. This gives $294 \times 33.1 = 9{,}731$ I.H.P.

Again, Table XI shows that the *Hopkins* gave 23.14 I.H.P. per revolution; and, for one engine, at the rate of 24.04. This was with 252 pounds steam and .720 cut off. With steam at nearly full pressure and .743 mean cut off, as on the official trial, we certainly realized 24 I.H.P. per revolution. Increasing 23.14 in proportion to the cut off and absolute steam pressure gives more than this, but the increase from either change is not in direct proportion.

At 24 I.H.P. per revolution, run 8, Table IX, had a horsepower of $24 \times 390.5 = 9{,}372$. This is the foundation of the dot-and-dash line in Fig. 8.

The corresponding speed for 9,731 I.H.P. is 30.1 knots, and we think it absolutely safe to predict that 30 knots could be realized. The required revolutions are $9{,}731 \div 24 = 405$, and the required pitch of propeller with, say, 9.5 per cent. slip, is 8.22, or, say, 8 feet 3 inches. With 42.5 square feet of projected area the thrust

per square foot would be 2,261, so that propeller Z slightly fined in pitch should be satisfactory.

Of course, diminishing the H.P. cut off to make a more efficient engine, and fining the propeller, would give a trifle higher result; but the piston speed becomes very great, and shortening this cut off would soon make torsional vibration reappear.

Conclusion.—In conclusion, we are happy to be in the position to say that never in any of the trials had we any accident to any one aboard, or were we held up by any bearing getting hot, with the doubtful exception of the first trial of the *Hopkins*, 28th June, 1902, when two pieces of steel, which should never have touched, threatened to fire. This is only one indication of the great excellence and faithfulness of the crew which ran these trials, almost all of whom were there from first to last. We regret that Mr. S. S. Caskey had to withdraw his valuable services, before the completion of the *Hopkins* trials, to attend to urgent personal interests.

We are greatly gratified that the high reputation of the builders has been fully maintained in these boats. In spite of the heavy overweight she had to carry, the *Hopkins* is one of the few destroyers, authorized at the same time, which have reached their original contract speed, and one of the very few which have considerably exceeded it.

EXTRACT FROM ADMIRAL MELVILLE'S PAPER ON THE VIBRATION OF STEAMSHIPS. "ENGINEERING," VOL. LXXV, P. 303.

"EXPERIENCE WITH, AND CURE OF, UNITED STATES TORPEDO-BOAT DESTROYER *HOPKINS*.

"Another very instructive case happened with the United States torpedo-boat destroyer *Hopkins*. At about 250 to 270 revolutions, the forward engine worked very roughly, and the ship vibration became somewhat severe. At higher revolutions this engine ran most satisfactorily; but at about 300 revolutions the after engine vibrated most dangerously, and this was accompanied by such excessive ship vibration that the captain on the bridge could not see the horizon.

"It seemed that something was holding the after engine back. Its designed revolutions for full power are 400 per minute, but with full steam of 270 pounds in the boiler it would not run more than a little over 300. After the changes were effected in the moving masses and the cut-offs of the after engine, the sister ship *Hull* ran as high as 419 revolutions per minute. At the time of writing the *Hopkins* has not been tried again.

"The following are the data from a run of the *Hopkins* at a little under full boiler pressure, before alterations:—

JULY 8, 1902.

.....	Forward engine.	After engine.
Steam by engine-room gauge...	224 pounds.	217 pounds.
Revolutions per minute.....	347	299
Indicated horsepower:—.....	% of total.	
High pressure.....	958	649
Intermediate pressure.....	1,376 } 2,334 = 63%	Not obtained.
First low pressure.....	680	Not obtained.
Second low pressure.....	677 } 1,357 = 37%	424
Total.....	3,691 Diff. = 26%	Between 2,000 and 2,500
Total revolutions in one nautical mile.....	884	762
Speed of ship in knots.....	23.5	
Slip of propeller taking 8.5 feet pitch.....	19%	6%

"The propeller was not a true screw, and therefore a definite pitch cannot be stated.

"An examination of this result is very instructive.

"The indicated horsepower was only obtained for two cylinders of the after engine. The vibration was so bad that even this was difficult to get. But the total indicated horsepower, by comparing the results from the two engines, is seen to be certainly over 2,000, and probably under 2,500.

"The smallness of the indicated horsepower in the after engine is not all accounted for by the smaller revolutions and steam pressure. It seemed to indicate somewhat leaky valves. This was verified on examination. But the very remarkable feature is the great difference of revolutions and slip.

"If there was no wake—and here the run to the propellers is very clear—the horsepower of each engine, used in propulsion, would be roughly proportional to the square of the slip; that is, to the energy imparted to the column of water in which the propeller works. Hence, if the action of the forward engine is normal, and if the efficiencies of the forward and the after engine were equal, this comparison gives for the after engine only

$$3{,}691 \times \frac{6^2}{19^2} = 368 \text{ I.H.P.}$$

"(Allowing for the wake, this figure would be rather larger.) Thus it appeared that nearly 2,000 indicated horsepower was being wasted.

"From this it was first concluded that the propeller of the after engine was bent; and in any case the vibration was so great that a thorough examination was imperative. The ship was therefore docked after two single runs on the mile at such high speed, but the condition of the propeller was found to be perfectly satisfactory. A beginning of the notching, to be mentioned later, was alone discovered.

"After considering all possible ways in which loss of power could be accounted for, there was still no doubt that it was wasted at the propeller. It can readily be shown from equations 10a and 11a of Appendix I that the power absorbed by transverse vibration of the ship is exceedingly minute in amount. Internal friction of the engine is, of course, entirely inadequate in this case; besides, the engine turned very freely, and ran without a sign of heating. On the other hand, with great torsional vibration the resistances acting on the propeller are such that the power absorbed may rise to be large. Thus it became very probable that the whole trouble arose from torsional vibration of the shafting, and a calculation similar to that given above for the *Lawrence* showed that the elastic vibration of natural period should occur at the rate of about 540 per minute for the forward and 618 for the after engine. This gave the critical revolution for the forward engine as 270 and for the after as 309 per minute, which practically agreed with the trial.

"The arrangement of cylinders and cranks is the same as for

the *Lawrence*, and the same means was taken to reduce the second-period impulses due to steam and inertia.

"The alterations were first completed on the sister ship *Hull*, and no difficulty was experienced in running at the critical revolutions; though, as in the *Lawrence*, the engines seemed to make a little more noise at these speeds. As it seemed injudicious to alter the cuts-off in either boat so much as to get a nearly complete balance for second period, it was intended to use the by-pass into the low-pressure receiver to help to bring up the power of the low-pressure cylinders. Using this on the forward engine of the *Hull* seemed distinctly to diminish the noise at 250 to 270 revolutions, though at all speeds the running of the engine would have been judged excellent. Perhaps the most remarkable evidence of the change in mechanical efficiency is given by the fact that while the *Hopkins*' after engine ran at only little over 300 revolutions, with 270 pounds steam, the initial pressure in the high-pressure card of the *Hull's* after engine, subsequent to the alterations, was only 130 pounds for 318 revolutions, all other conditions being the same as the *Hopkins*. The lengthening of the cut-off accounts only for about one-fourth of the reduction of initial pressure.

"The percentage distribution of power in the *Hull* at 1,145 indicated horsepower and 231 revolutions was about the same as that given for the *Lawrence*, but was rapidly improving as the power went up; and this occurs also with *Lawrence*. It may be a matter of surprise that so much improvement was effected when, even at critical revolutions, considerable unbalance remained. But,

"1. When the cause exciting vibration is reduced so much that there is no back-lash or jar, the destructive power of the action is at once immensely mitigated. Wöhler's well-known experiments show that the stresses ranging up to 5.40 tons per square inch, quoted from Frahm's paper, would be in no way dangerous; but severe vibration must usually induce serious jarring. For instance, in the *Hopkins*, not only was the noise in the engine room deafening, but at the stern the slamming of the shaft in the stern bearings at the critical revolutions was almost unnerving. This has entirely disappeared in the *Hull*.

"Frahm shows that the amplitude of torsional vibration is proportional to the exciting couple, but his estimates of stress necessarily take no account of slack or jarring.

(Reducing the exciting forces of transverse ship vibration to half is far from reducing the effect in the same proportion. This will be evident after reading Appendix I. Hence the balance to prevent transverse vibration must be much more perfect than in the case of torsional vibration.)

"2. The rate at which a propeller would damp torsional vibration must diminish greatly when the slip gets small and the blade is moving approximately 'edge on' to the water. So much power was being absorbed in the *Hopkins* that the after engine dragged behind the forward one, reducing the slip. This, in turn, reduced the rate of absorption for a given amplitude, and thus allowed the amplitude to rise. If steam had been partly shut off the forward engine to make them run together, there seems little doubt that the vibratory action would have been much mitigated. Hence, when the balance was so far effected that the engines in the *Hull* ran together readily, the good effect of the alteration was fully realized.*

* [Note added February 7, 1903.—The statement in the preceding paragraph seems to be remarkably confirmed by the trials of the *Macdonough*, sister ship to the *Lawrence*, which have been made since the foregoing was written.

The second-period torsional balance of the after engine of the *Macdonough* is slightly worse than that given above for the *Lawrence*. But so long as this engine ran faster than the forward engine, the result was satisfactory. On run 2, of December 2, 1902, we have:

.....	Forward engine.		After engine.	
	High + intermediate cylinder.	Two low-pressure cylinders.	High pressure + intermediate pressure cylinders.	Two low-pressure cylinders.
Per cent. of total power..	55	45	59.7	40.3
Revolutions.....	320		332	
Indicated horsepower...	2,820		2,528	
Power per revolution....	8.81		7.61	

"FALL OF PROPELLER EFFICIENCY.

"Frahm remarks (translation, page 743): 'Another important side of this question lies in the fact that not only the efficiency of the engine is reduced by the vibrations which I have found on the *Besoeki*, but also the propeller efficiency suffers considerably by the great variation of speed, a result which is obtained by several observations and investigations, too lengthy to be considered here.'

"The great waste of power noted for the starboard engine of the *Hopkins* indicated a propeller efficiency lowered in a remarkable degree.

"The same effect is exhibited by the *Lawrence*, but in a much less degree, even after the partial balance had been effected. This is shown by the following from a letter of Mr. Chas. B. Edwards, superintending engineer of the company which built this boat: 'It was quite noticeable that the after engine had more vibration and labored harder from 320 to 360 revolutions, and for the same steam pressure ran from 10 to 18 revolutions

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The after engine is near its critical revolutions; and, especially as the power per revolution is lower for this engine, the elements of trouble are markedly present should the after engine run slow.

For some reason, not yet precisely ascertained, the after engine did run slow on the next trial—January 4, 1903. As the after engine approached its critical revolutions, vibration became somewhat severe, and it fell further and further behind the forward engine, till on runs 7 and 8, at 235 pounds steam for both engines, the mean revolutions and slip were:

	Forward engine.	After engine.
Revolutions per minute	376.6	332.9
Difference of revolutions	43.7	
Speed in knots	27.72	
Slip per cent	21.5	11.2

If this explanation is true, the trouble will be readily obviated by equalizing the power per revolution and further improving the balance. Should the result be known before these articles are completed, it will be noted.]

The trouble was readily obviated as suggested above, and by forcing the engines to run together. The *Macdonough* has since successfully completed her official trials.—J. H. M. AND A. C. L.

per minute less than the forward engine. On our 29.4-knot run the forward engine averaged 370 revolutions, while the after engine averaged 359 revolutions per minute at the same steam pressure.'

"The strong impulsive action of the vibrating propeller will often be sufficient to raise the fluid pressure to that at which cavitation will be produced. This cavitation will be intermittent, the cavity forming at the backing side of the blade, near the following edge, each time the direction of the vibratory rotation coincides with that in which the engine is driving the shaft. Thus, in the *Hopkins* a cavity was no doubt formed twice per revolution. If we consider the rounded form of the backing side of the blade, it is evident that the complete or partial annulling of the water pressure by the formation of a cavity in the position stated will cause the total resultant pressure on the propeller to be less directly astern. This longitudinal component is the one which drives the ship; but as it has no moment, resisting the rotation of the propeller, it does not absorb any power of the engines. The component normal to this alone absorbs power. "Cavitation or strong eddying by increasing the latter in proportion to the former will thus clearly reduce the propeller efficiency. The foregoing may appear to be a paradox, as the engine power is principally absorbed by the resistance of the ship. But as neither component of propeller resistance—the efficiency being above zero—can exist separately, there is no mystery.

"The accelerated scour of the water over the propeller surface would also slightly lower the efficiency by affecting the resultant forces in the same way as cavitation; but probably this is a relatively unimportant action.

"DISINTEGRATION OF PROPELLERS.

"It has been frequently found in United States torpedo boats that after running for a time a notch forms about two-thirds of the radius out from the center on each blade. Many fanciful explanations have been offered, and Macalpine guesses that it is due to torsional vibration. It is shown clearly in Figs. 47 and

48, which are from photographs of the United States torpedo boat *Stringham*. The vibration is more or less like the crack of a whip. If we conceive a blade with no pitch, this vibratory action would alternately put a strong compression and extension on both leading and following edges of the blade. The resistance of the water, on the other hand, would put a tension on the leading edge and a strong compression on the following edge. Thus, if the effect were strong enough, the leading edge would be stretched, while the following edge would be compressed; and if it tapered off sharply enough, the bronze would buckle as shown. Now, in an actual blade this buckling action could not happen near the root, as the acting forces would bend it about the long axis of the root section; but when we get out to where the notching actually occurs we have a case not dissimilar to that of the blade with no pitch. The notching always occurs on the after edge, as this explanation demands; and the bulge seems always to be aft, as Fig. 47 shows, and seems most probable from the supposed action.

"When the *Stringham* was docked it was found that a large area on the aftermost, commonly called the 'go-ahead,' face of the blades, extending nearly half-way across the blade from the outer half or two-thirds of the leading edge was scoured bright, while the rest of the face was dull and overgrown with sea grass. This is not well brought out by the photographs. A similar effect was shown on the other, or 'go-astern,' face of the blades, but the bright part was near the following edge. The bright parts are exactly where we would expect the greatest scour. By the time the water got well across the after side of the blade there would be a layer of sensible thickness which had little motion relative to the surface of the blade, thus mitigating the scouring action. On the forward surface cavitations or strong eddying would be sure to produce a powerful scour near the following edge.

"Experience with the *Hull* tends to support the foregoing explanation of origin of the notching. She was not run till after the changes, partially balancing the tortional impulses, had been made. When docked, after being run at full power, all the blades

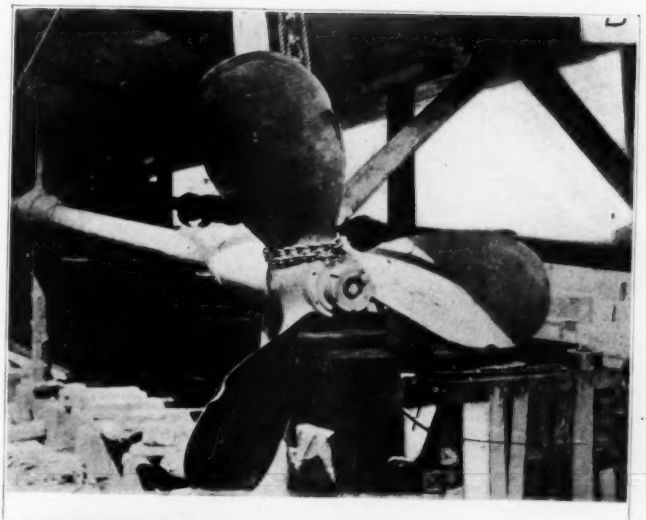


FIG. 47.

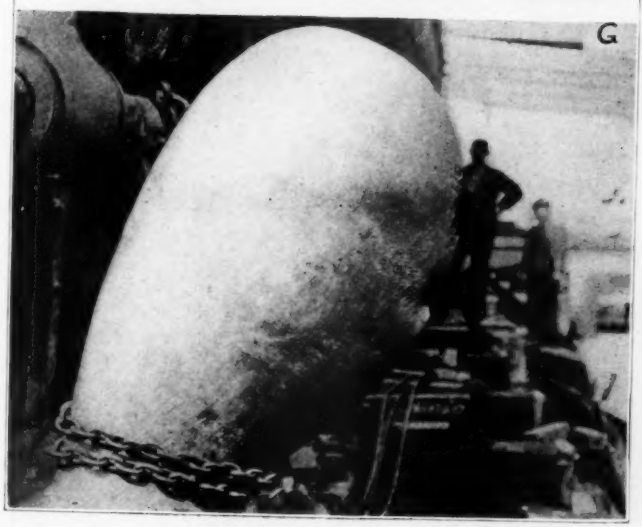


FIG. 48.

were perfectly free from this action except one on the starboard propeller (that driven by the after engine), in which just a trace of it was visible. The *Hopkins*, with the scour design of propeller, had shown it quite markedly.

"I reserve my judgment, but if this is the true cause, it shows that severe torsional vibration is not uncommon."

CARE OF TORPEDO BOATS IN RESERVE.

ATTENTION AND OPERATION ESSENTIAL FOR EFFICIENT MAINTENANCE.

By Lieutenant G. C. Davison, U. S. N., Member.

The following notes were written at the request of the Council of the American Society of Naval Engineers, with the hope that they may be of some practical assistance to young line officers detailed to duty in connection with the torpedo boats in reserve who have had no previous experience with either torpedo boats or destroyers.

COMPOSITION OF THE FLOTILLA.

The reserve torpedo-boat flotilla has its headquarters at the Navy Yard, Norfolk, where there has just been completed an admirably designed system of slips for the berthing of the boats.

The flotilla consists at present of the following boats: *Bagley*, *Barney*, *Biddle*, *Cushing*, *De Long*, *Dupont*, *Ericsson*, *Foote*, *Mackenzie*, *Porter*, *Somers*, *Stockton* and *Thornton*. The *Rodgers* is officially classed as within the flotilla, although for the past year she has been detailed for special duty in connection with the extended series of tests that have been conducted for determining the value of liquid fuel for naval purposes. For practical purposes her status has been that of a boat in commission rather than that of one in reserve.

Regarding the reserve flotilla as a single unit, it corresponds to a vessel of the following description:

Displacement (combined tonnage of all the boats)	2,175
Number of boilers	32
sets of main engines	26
cylinders of main engines	101
auxiliary appliances	244
Total horsepower	38,670

Stated in another form, the flotilla of torpedo boats has practically an installation of horsepower equal to that of the most powerful ocean greyhound afloat on a displacement of a vessel about the tonnage of the U. S. S. *Detroit.*

VARIED DESIGNS OF CONSTRUCTION FOUND IN THE FLOTILLA.

The boats of the flotilla not only differ greatly in tonnage, but also in the design of the motive machinery. There are found among them representatives of almost every type of torpedo boat in existence. As for the design of the motive power of the several boats, the following installations of engines are found: three-cylinder triple-expansion; four-cylinder triple-expansion; four-cylinder quadruple-expansion, and five-cylinder quadruple-expansion.

Two of the boats have single screws, the rest having a twin-screw installation of motive power. Of the two single-screw boats, one has a right-handed screw, while the other has a left-handed screw. Of the twin-screw boats, three have in-turning screws and nine out-turning screws.

The types of boilers represented in the flotilla are the locomotive type, three designs of Normand, two of Thornycroft, and one of Mosher. The boiler installation is thus sufficiently varied to note their relative value as to efficiency and endurance.

There is a great variety of design as to the manner in which the auxiliaries have been installed. Considerable latitude was given the various builders in the installation of auxiliary machinery, and the advantages and disadvantages of special installations have thus been noted.

VALUE OF EXPERIENCE WITH DIFFERENT DESIGNS OF TORPEDO BOATS.

There must of necessity be a difference in the efficiency and endurance of the several designs of boats, and probably this difference can best be noted by those who are entrusted with the care and maintenance, as well as operation, of the several types.

The relative worth of the several types can undoubtedly be definitely ascertained by carefully noting the behavior of the boats under similiar conditions, and by keeping a record of the repairs required.

The tests to which the several boats are subjected are sufficiently varied to determine the endurance and efficiency of the rival designs. Such observation also ought to furnish the Department definite information as to the character of the workmanship that was put upon the boats by the several shipbuilding establishments. In all probability no other Navy has collected a flotilla of torpedo boats of such varied character, and as these boats receive like care and attention, and are handled by the same crew, their defects and merits should be soon manifested. It is certain that however inclined individual officers may be to favor certain types of boats when joining the reserve squadron they lose such bias when comparative experiments show the superiority of another design.

While it might be to the interest of the naval service to possess fewer types of torpedo boats, it certainly adds to the interest and benefit of individual officers to experiment with the different kinds. In the training of machinists, water tenders and oilers, it is of great advantage to possess this variety in installation of motive power; for after either officer or enlisted man has been connected a year or eighteen months with the reserve flotilla boats, he will not often find much that is new to him in the care or operation of any kind of torpedo craft.

PRIMARY PURPOSE OF ESTABLISHING THE RESERVE FLOTILLA OF TORPEDO BOATS.

The object in forming a reserve torpedo-boat flotilla was to maintain the boats in such a state of efficiency that not only could they be put in commission with a few days' notice, but that they would be actually able to perform at sudden call the military functions for which they were built.

While torpedo boats can be kept in the highest state of efficiency by keeping them in commission in actual service, when one remembers that in all navies these boats have been used to carry the mail, and for all manner of auxiliary duty not connected with the purpose for which they are built, it may be that keeping them in reserve is an excellent way to maintain efficiency.

In regard to the training of the personnel, the best results can

undoubtedly be secured by keeping the boats in commission, since a greater number of men can be trained in that manner.

IMPORTANCE OF SKILLED MEN IN CHARGE OF THE FLOTILLA.

It should ever be kept in mind that there is nothing very difficult nor mysterious about the installation of the motive power of a torpedo boat. The main engines and auxiliaries, however, are lighter, more compact, and have a higher piston speed. Measured by the horsepower developed, there are fewer men to look out for such machinery while it is actually in use. The following comparison will show this disparity: The horsepower developed by a first-class torpedo boat of 180 tons displacement and of a cruiser of 2,500 tons is about the same, that is, about 4,000 I.H.P. In both types of war vessels the same amount of coal is burned and the same amount of water evaporated to develop this power. The engineer force of the cruiser probably consists of fifty men, while there are only fifteen on the torpedo boat. While the work on the cruiser is more continuous, the work on the torpedo boat is more exacting and trying. The crew of the torpedo boat should likewise be more resourceful and skillful, and capable of enduring, at least for a time, more severe and distressing work. Probably every skilled machinist, water tender, oiler or fireman could do efficient work on a torpedo boat, but it would be absolutely essential that such men should be given sufficient time to become familiar with their new surroundings and duties. Even a picked crew from a large vessel would bring to grief the machinery of any torpedo boat if they attempted to take the boat to sea a few days after joining her, providing they had had no previous experience with torpedo boats. If the same picked body of men had two or three weeks of systematic training and instruction, they might be depended upon to do efficient work.

In the care of torpedo boats in reserve it is essential that the leading men, both on deck and in the engine room, should possess skill, intelligence and endurance. It is these men who set the pace for the remainder of the crew, and in making up the composition of the flotilla this fact should be kept prominently in mind.

ORGANIZATION.

In the efficient maintenance and operation of a dozen torpedo boats kept in reserve, organization and system are the first requisites for their proper care and endurance. Without such organization it would be impossible to keep the boats in uniformly good condition. It will be found that on each of the boats there are important duties that are likely to be neglected, and only a thorough system of organization will secure the continuance of the necessary work. It is likewise essential to see that the reserve crew takes an equal interest in the maintenance of all the boats, for it is but natural that undue care and attention will be given special work unless general interest is manifested in all the duties assigned each enlisted man.

There has been allowed for the efficient maintenance of the torpedo-boat flotilla three commissioned officers, two gunners, a complete crew for one boat, a skeleton crew of six men for each of the other boats, and a complement of petty officers sufficient for the general work of the flotilla in the care of the various military and engineering auxiliaries installed on the several boats of the flotilla. The full crew allowed for one boat has been assigned to the *De Long*, and consists of

1 Chief gunner's mate,	2 Chief machinists,
1 Gunner's mate, 1st class,	4 Machinists,
1 Gunner's mate, 2d class,	4 Water tenders,
1 Quartermaster, 1st class,	4 Oilers,
1 Quartermaster, 2d class,	4 Firemen, 1st class,
1 Ship's cook, 2d class,	1 Cook,
1 Steward,	1 Mess attendant.

The following petty officers are allowed for the general work of the flotilla:

1 Chief yeoman,	1 Electrician, 1st class,
1 Yeoman, 3d class,	1 Blacksmith,
1 Carpenter's mate,	1 Boilermaker,
1 Painter,	1 Coppersmith,
1 Ship's cook,	2 Landsmen.

In addition there is allowed each boat, except the *De Long*, a reserve crew as follows:

1 Gunner's mate,	1 Machinist,
1 Oiler,	1 Machinist, 2d class,
1 Coal Passer,	1 Water tender.

CHARACTER OF ROUTINE DUTIES PERFORMED.—GENERAL DUTIES CARRIED ON.

From various causes the complement allowed is seldom filled. At the present time coal passers are doing the duty that should be done by water tenders and oilers, but as these men are anxious to learn, they are doing fairly well, although it entails much extra work upon the part of officers to instruct them in their duties.

The six men detailed to each individual boat are required to live in her, and it is exacted that at least two shall remain on board every night. The machinist looks out for an engine room with all its auxiliary machinery. The oiler is responsible for the condition of affairs in the other engine room. The water tender looks out for both fire rooms. The gunner's mate is responsible for everything outside of the engineer's department. These six men are required to move every engine and auxiliary daily, keep bilges clean, scrape and paint metal work where necessary, and make minor repairs. After a torpedo boat has been put in good condition, it is possible for the complement assigned each boat to do all routine duty before 9·30 A. M., at which hour all hands are mustered on the deck, orders are published and work assigned for the day. Then commences the innumerable repairs, improvements and alterations that are suggested by the experience with other boats.

It has not been found possible to handle such a large number of boats as a single unit, therefore the following system has been adopted after progressive trial and development. The boats are divided into two groups of six each, and the skeleton crews into two groups likewise. Thus in each group there are twenty-four men, the six skeleton crews of six each, a number sufficient for a full crew to run one boat for an extended period. Whenever a

boat is sent to sea for a run, one-half of the *De Long's* crew goes with that boat, the *De Long's* complement being a picked one, whose duty it is to instruct and train all new comers and inexperienced men. The remainder of the crew required for the run is taken from one of the two groups of skeleton crews mentioned above.

Each main group of enlisted men that is responsible for a group of boats is again divided into two contingents, the deck and engineer's force. The deck force is in charge of a chief gunner's mate, and works together, after 9·30 A. M., as one unit upon the particular boat that is in need of attention or repair. The engineer's force of each group works under the direction of a chief machinist, and is about as large a working party as any one chief petty officer can efficiently handle. The fact that a general working party comes on board does not relieve any individual responsible for the cleanliness and efficiency of any portion of a single boat of such particular duty. In general, it has been found that the individuals become very much interested in the work assigned them, but in some cases this pride is manifested in highly polishing bright work, while pistons and valve rods, slides and other working parts, which should be kept scrupulously clean, are likely to be neglected. It is of advantage, however, to create a spirit of emulation upon the part of individual men, for it is often the case that hints and suggestions are made by the enlisted force which are of great value.

CONCENTRATION OF FORCE UPON ONE BOAT AT A TIME.

Considering only one group, each torpedo boat of the group is thus taken in hand by the group of men responsible for her condition, and work is continued on her until she is put in the highest possible state of efficiency. In the conscientious discharge of this work both officers and men derive great satisfaction. When such work is completed the boat is turned over to the special detail assigned to her, and that contingent is held rigidly responsible for her continued efficient maintenance.

Each boat, after being put in efficient condition, is subjected to an actual steaming trial at sea, and as all connected with the

reserve flotilla understand that the highest efficiency must be secured, everything is put in shape to meet this contingency, which occurs once per month. In carrying on such tests the following work is done:

Preliminary Details to be Attended to.—1. Boilers are closed, filled to steaming level, furnaces primed, and everything put in condition for undergoing continuous forced-draft trials.

2. The main engines and auxiliaries are jacked one complete turn. Valves are tested, and particular care taken that no loose tools or articles are lying around.

3. Care is also taken that the necessary stores and tools requisite for temporary repairs are placed on board the ship.

4. A personal inspection is made by the chief machinist in charge to see that all nuts, lock-nuts and split pins are properly set up.

5. The drainage system is carefully examined and strainers are cleared. All who are to go on the steam trial repair to their stations and thoroughly familiarize themselves with the location of every valve and appliance that is essential to the operation of the machinery.

While Under Way.—1. During the steam trial the working of every running part is carefully noted. All defects, such as leaks, knocks and unsatisfactory working, are carefully recorded.

2. Water from the condensers and boilers is carefully tested.

3. When clear of the dock, the machinists, second class, are stationed at the throttle and instructed in handling the engines. Signals are given suddenly and repeatedly to keep the men on the alert.

4. By reason of the engines of some of the boats turning inboard and others outboard the machinists at the reversing lever are instructed and drilled to take the position of the reversing lever as a guide to noting the direction in which the engines are turning.

5. In order that those stationed at the throttle can quickly approximate as to the speed at which the engines are turning, a series of trials is made, during which the revolutions are carefully ascertained and the corresponding receiver pressures noted.

From these observations a table is made up which is stamped on a brass plate and put up in the engine room where the man at the throttle can easily see it. By simply noting the pressure in the intermediate receiver, the engine speed can be thus quickly determined.

After the Steaming Trial.—1. Steam is kept on the boiler for pumping purposes, blowing tubes, charging torpedoes and for cleaning bilges. When this work is finished fires are allowed to die out.

2. Engines are cleaned while warm.

3. The drainage system is examined, and bilges cleaned, scaled and painted where necessary.

4. Fire rooms and furnaces are cleaned, scaled and painted.

5. Cylinder covers are removed, walls cleaned and vaselined, and covers put back. The same care and attention is given to the main engine valves.

6. Boilers are overhauled and completely filled with fresh water.

7. All impairments noted on the run are corrected.

8. An official report is made to the commanding officer, showing that the detailed requirements have been carried out.

OVERHAULING AND EXAMINATIONS CONDUCTED EACH QUARTER.

(*a*) Boilers are cleaned and washed out thoroughly. All attachments are examined and zinc plates renewed where necessary. They are then closed and completely filled with fresh water.

(*b*) Pistons, piston rings and valves of main engines are removed, cleaned, vaselined, replaced and adjusted.

(*c*) Condensers are opened and an occasional tube is removed to note their condition. The interiors are cleaned when necessary.

(*d*) In order to eradicate grease and thus prevent both deterioration and impairment, the boilers and condensers are boiled out if there is any sign of such coating.

Where there has been good construction, and where the boats have previously had excellent care, this cleaning up and overhauling can be done in one or two days. Where there has been

weakness of design or lack of care and attention, this repairing and overhauling may occupy a week. Whether the overhauling requires one day or several weeks, the skeleton crews thoroughly understand that work must be continued on each boat until everything is in a state of efficiency. The enlisted men are thoroughly appreciative of the fact that when things are put in good condition the boat will give very little trouble afterwards. It is certain that if the overhauling is not done in an efficient manner defects and weaknesses will develop on the next run, and thus this lack of attention will manifest itself.

RECORD OF WORK CAREFULLY KEPT.

There is kept on board each of the torpedo boats a record book wherein is noted all the work that has been done upon hull, machinery and ordnance appliances. There is a special page for each piece of auxiliary machinery, wherein can be noted the item of work, by whom performed, number of hours employed and condition of the appliance.

At noon each Saturday the two chief machinists hand in a report of the work done during the week. From these reports the record book of work is made up. It is absolutely essential to devote care and attention to keeping up this record of work, since there are several hundred pieces of machinery on board each boat to keep track of. This record book of work serves a twofold purpose; first, it shows when each piece of machinery was overhauled; second, it serves as a check upon the careless or indifferent, for as long as this record of work is kept, it is not likely that neglect or inattention to important appliances can continue for any length of time.

IMPAIRMENTS AND CASUALTIES WHICH ARE LIKELY TO OCCUR.

Boilers.—The boilers of a torpedo boat can be easily impaired, and this fact is repeatedly impressed upon those stationed in the fire room. The fire-room complement are likewise distinctly made to understand that their greatest safety lies in being attentive to duty, ready in resource and cool in emergencies.

Most of the boilers on the torpedo boats are either of the Nor-

mand or Thornycroft types. There are some of other designs, but they all may be treated as belonging to one of the two general classes named. These boilers are built for the purpose of obtaining a maximum evaporation with a minimum of weight. The tubes are generally about one inch in diameter, as in a boiler of given weight and dimensions the smaller the tubes the greater the heating surface. While these tubes when new are sufficiently strong to withstand many times the pressure carried in the boilers, their original thickness is so light that but little margin can be allowed for corrosion. When deterioration of the surface of the tubes commences impairment proceeds very rapidly.

While these boilers are light and small, they are still expected to develop excessive horsepowers, and this means excessive evaporation from the heated surfaces. In consequence, when one of these boilers is subjected to forced-draft conditions, the circulation of water in the tubes must be very rapid and complete. When forced to the utmost, the entire contents of a torpedo-boat boiler is evaporated every ten minutes, and therefore some of the tubes are likely to be burnt out if the supply of feed water is stopped for even a few minutes.

Overheating of the Tubes.—The overheating of the tubes and the resulting leakage of steam and water, sufficient in many cases to endanger the fire-room force, can either be ascribed to low water in the boilers or to exceedingly defective circulation. The causes that will produce low water are many, but the following are the principal ones:

1. Choking up of the fittings or glass of the water gauges.
2. Defective design of feed arrangements.
3. Ignorance or negligence upon the part of the water tender.

In all probability it is either ignorance or negligence that are responsible for ninety per cent. of the mishaps to the boilers of torpedo boats.

The following are a few instances of the many that could be given as to how the boilers of naval ships are impaired:

(a) About six years ago a torpedo boat was lying alongside a wharf in the winter time with steam on one boiler for heating

purposes. During the night, when an experienced water tender was on watch, water got low in the boiler. The top drum was overheated, and all the tube ends were loosened in the top drum. It was found out later that every arrangement for feeding the boiler was in perfect order. The experienced water tender had simply gone to sleep.

(b) On another occasion (not on a torpedo boat) steam was gotten up one morning on a boiler and the vessel got under way. A few minutes later the steam pressure went down to zero, and it was found that the boiler tubes over the furnace were a mass of molten metal. Yet all the time the gauge glass had shown the proper water level. The cause of this casualty was that the night before some one had closed the valves on the water column and they had not been looked at nor opened in the meantime.

(*c*) On another craft with water-tube boilers there was an inexperienced machinist on watch in the engine room, and a very stupid fireman in the fire room. These men were stationed there for the reason that better ones could not be obtained. There was some trouble with the feed pump, which was in the engine room. The fireman saw the water disappear in the gauge glass and quietly remarked to the machinist that "he thought the water was getting low." Neither man seemed to have regarded the situation as serious, and allowed events to take their natural course, without calling any one to assist them. In a few minutes the steam pressure went down, the engine stopped, and there were two burnt-out boilers.

Many more instances could be given where either ignorance or negligence has been the primary cause of marine boilers being injured, but the fault is not always with the firemen. The experience of the past five years with automatic devices for regulating the supply of feed to the boiler ought to show that such appliances are absolutely unreliable and uncertain, for under actual sea conditions the presence of muddy water, lumps of grease or expansion of parts would interfere with efficient working of the device.

Priming of Boilers.—When boilers prime continuously or vio-

lently one may look for all kinds of trouble. Cylinders are likely to be cracked, connecting or piston rods bent, pumps disabled or pipes burst. Some machinists do not seem to realize the possible danger resulting from this source, and it is therefore necessary repeatedly to caution them as to the effect of permitting priming to continue. The principal causes of such priming on torpedo boats are:

(*a*) Faulty design in the arrangement of baffle plates, as well as of feed and dry pipes.

(*b*) The failure to clear the boiler of either grease, mud or soda.

(*c*) Excessive amount of salt feed.

(*d*) Improper regulation of valves.

(*e*) Irregular firing upon different portions of the grate.

Where defect in design is responsible for priming, experiments should be made to determine if the evil could not be decreased by either reducing the amount of coal consumption or by change in the arrangement of the feed.

Where the tubes are covered with grease or mud it will be found that careful and repeated cleaning and washing out of the boilers will greatly improve matters.

Priming has often been stopped by reducing the opening of the stop and throttle valves, or by changing the size of the steam pipes.

Where irregular or improper firing is done there must, of necessity, be an interference with the natural circulation, since there cannot be a uniform heating of the tubes under such circumstances.

Deterioration of Tubes.—The small, thin tubes of torpedo-boat boilers deteriorate very rapidly unless special attention is given to the care and preservation of the boilers. From various causes these tubes are attacked both on the interior and exterior surfaces. Exceeding care should therefore be taken in the inspection of boiler tubes. One needs only to have had but little experience in the operation of torpedo boats to realize that something else is required of a naval inspector of boiler tubes than to conduct a few physical tests, and to take care that they are of proper gauge and length.

The presence of grease in such a boiler will soon cause some of the tubes to leak, for, by reason of the decomposition of the grease, the metal is attacked, and once corrosion sets in it will not be long before holes will appear in the tubes. The grease is also a non-conductor, and thus the efficiency of the boiler is decreased by its presence. In order to prevent either the drums or the tubes from being coated with grease, care should be taken that no oil is used in any steam cylinder. The piston and valve rods only should be lubricated. The boiler should be boiled out at intervals, and washed out often, for with boats in reserve there will always be an unlimited amount of fresh water available for cleaning purposes. The water should also be tested weekly with blue litmus paper to see if there is any acid reaction. A few pounds of soda added to the water in the boiler prevents acidity; but where soda is used the thorough washing out of the boiler must follow.

Corrosion of the exterior of the tubes is most liable to occur around the ends. In order to prevent such deterioration the smoke-stack covers should be kept on when the boiler is not in use, for a single rain shower may indirectly do a great amount of harm if the covering is not put on the stack when fires are not lighted. Ashes are also likely to collect around the lower ends of the tubes where they are expanded into the lower drums. If ashes are not removed, and become moist, they will harden and cake. It will then become quite difficult to remove the deposit, and it only requires a little additional salt water to set up a chemical action which will, in a short time, cause many of the tubes to leak and thus put the boiler temporarily out of commission.

It is intelligence and continuous care that is requisite in the maintenance of torpedo-boat boilers. Such boilers may possess endurance when properly looked out for. They become exceedingly tender when neglected.

Skill Requisite in Firing.—Firing on a torpedo boat is an art. It is a comparatively simple matter to draw up specific directions regarding the thickness of fires to be carried, the size of coal to be used, the height of water in the boiler to be maintained, the

air pressure in the fire room, the manner of cleaning fires, and other details, but proficiency only comes from experience, judgment and resource.

Although even the skilled water tender may do very well on a torpedo boat to which he has been accustomed, such a petty officer upon going to another boat may have his steam pressure varying within wide limits until he learns just how things should be managed. If such a man, however, is permitted to do duty for several months on boats of a reserve flotilla, he will be able to adapt himself in a day or two to forced-draft conditions on any kind of boat.

If it could only be generally known how seriously efficiency and endurance is dependent upon skilled firing and good water tending, then the water tenders would be given pay and position commensurate with the important duties which they perform. Compared with the duties of a fireman, the work of the water tender is becoming more important each day. In many instances a skilled water tender not only does his own work, but does that of a first-class fireman also, and yet there is only a difference of a few dollars in the monthly pay of the two ratings. There is probably not an officer connected with any torpedo-boat flotilla who is not of the opinion that something special should be done not only to secure a full and competent complement, but also a reserve of water tenders.

CASUALTIES AND IMPAIRMENTS TO MACHINERY.

In the repair and operation of torpedo-boat machinery high-grade men must be detailed for duty in the engine rooms. This is not a station where commonplace petty officers can do efficient work. Special requisites are essential for the man at the throttle, for he must not only be alert in action, but he must also have manual skill, readiness of resource and an inclination for work. But however strong the complement in the engine room, casualties and impairments may occur through carelessness, and therefore all are enjoined to keep everything in the highest state of efficiency to ward off mishap or danger.

The causes of mishaps to torpedo-boat machinery may be divided into the following classes:

1. Faulty design.
2. Defective construction.
3. Untrustworthy material.
4. Normal wear, gradual deterioration, or unavoidable accidents.
5. Carelessness, ignorance, or neglect upon the part of the personnel.

In general, it may be stated that it is exceedingly probable that ignorance or neglect is responsible for as many casualties as all the other causes combined.

On account of the high rate of piston speed and consequent wear on bearings, the clearances in the lower ends of the cylinders of torpedo boats are decreased much more rapidly than in those of larger vessels. On board vessels which do a great amount of steaming, and where it has been necessary frequently to adjust the bearings, the men in charge are directed to note carefully the amount of cylinder clearances, for these clearances are not designed any too large.

Indicator attachments should be made to all the cylinders. After a boat is first put in commission, and whenever conditions as to speed, weather and draught vary considerably, cards should be taken in order to have reference records for future comparison. Cards should be taken, if possible, every quarter. While some torpedo boats have no indicator attachments, it will be found that it is inviting risk and danger not to make arrangements for doing so, for the cards will not only serve as a check to inefficient distribution of steam pressures, but may act as monitors in preventing casualties.

Bearings.—All bearings should be kept in good adjustment. With fast-working machinery it is absolutely imperative to take up every knock that is noted, otherwise there will be heating of the brasses or even breaking of the shaft.

The crank pin and crosshead brasses are usually adjusted to a No. 32 B.W.G. wire lead. Many machinists prefer to make adjustments by "swinging" the bearing. While expert mechanics

are often able to adjust very successfully by this method, it should not be depended upon, but should simply serve as a check to see that the bolts were not set up too tight.

While the main bearings and thrusts give comparatively little trouble when they are properly looked out for, periodical examinations should be made. Careful attention should be given at all times to the details of all journals and bearings, such as seeing that the nuts are well set up, and that lock nuts and split pins are properly put in place.

Air Pumps.—The water ends of some air pumps are fitted with soft-rubber valves, and where such valves are used repeated examinations should be made. The oil which necessarily reaches the cylinders from piston and valve rods after passing through the condenser reaches the channel ways of the pumps. This oil often lodges in pockets, and, upon decomposing, attacks the soft rubber. The metallic valves are liable to stick, due to their being gummed up with grease, and these valves should be examined and turned occasionally.

Irregular and unreliable delivery of water to feed tank is often due to the inefficient working of the detached air pump, and therefore both steam and water ends of this pump should be carefully looked out for, even to the minutest details.

Feed-Water Heaters.—As the feed water should reach the boiler at the highest possible temperature without impairing the vacuum, the feed-water heaters are an important element in securing efficiency of the boilers. They should be opened and cleaned out at least once every six months, or as often as grease accumulates in them. If the torpedo boats are not in use, however, the interval of cleaning may be of longer duration.

Forced-Draft Blower Engines.—These auxiliaries are subjected to harder work than any other appliances on board a torpedo boat. They are not only light, and run at excessive speeds, but they must necessarily be installed in positions where dust and soot reaches the running parts. On some torpedo boats as much as six inches of water pressure has been carried, and the engines have been speeded to a speed of five and six hundred revolutions.

It is exceedingly doubtful if such high forced-draft pressures

are either advantageous or necessary, for probably as much steam could be secured with less pressure provided skill and intelligence were exercised by the firemen in the stokehold. In consequence of the excessive strain and work to which the blower engines are subjected, it should be a rule to go over the working parts carefully after every run and take up all lost motion.

Removal of Grease.—In cleaning grease from the path of the steam cycle all the different parts of the system—boilers, cylinders, condensers, hot wells, feed-water heaters and feed pumps—should be cleaned at the same time. If this is done with care and judgment, it will be months before sufficient grease accumulates again to make it necessary to perform the operation, providing the only oil used is that on piston rods and valve rods, or that used in oiling the cylinder walls after inspection. It is surprising, however, that skilled men will be sparing of oil in the main cylinders, and yet expend it extravagantly in the small cylinders of the auxiliary appliances.

DOUBTFUL VALUE OF AUTOMATIC FEED AND STOP-VALVE DEVICES.

Many of the torpedo boats are provided with automatic boiler stop-valves and feed arrangements. While these appliances are very ingenious in design, and would probably work well when operated by highly paid and intelligent men, the average water tender and machinist are very skeptical as to their reliability and value.

It is thus found that upon many torpedo boats these devices are blanked off or disconnected, for the average enlisted man would rather rely upon operating the feed and steam valves himself than depend upon automatic appliances for regulating the control of either steam or water.

As a rule, the machinists and water tenders do not understand the principles upon which these devices operate, and, therefore, have no confidence in their utility. In the operation of every automatic device it is essential that grease, grit or any foreign substance be prevented from reaching any part of the mechanism, and as such conditions are impossible, the value of the device is discredited.

If an ordinary valve becomes stuck, there are mechanical means by which it can be loosened. In the case of automatic appliances such simple remedies will be unavailing, for it is generally requisite to remove the cover and to adjust special parts.

Probably some knowledge of the value of automatic devices may be gleaned by comparing the performances of the four similar monitors which have been officially tested during the past year. Automatic devices were fitted to the boilers of one of these vessels, and yet they were never used upon any of the official trials. No trouble was found in feeding the boilers of the monitors *Florida* and *Wyoming*, nor was foaming noted, and yet the steam generators of these ships were not fitted with a single automatic device.

ORDNANCE OUTFIT.

The guns and *small arms* of the boats of the flotilla are kept in the ordnance building in the navy yard, where they are cleaned and oiled once a week by a detail of gunners' mates from the boats.

The explosives and war heads, gun-cotton primers and ammunition are kept in the naval magazine at St. Julien's Creek, where the required tests and inspections are made by the personnel of the magazine station.

The torpedoes and their accessories are kept on board. Each torpedo is fired once per month. There may be a question as to the manner in which the torpedo practice should be conducted, as there are two ways. One way is to treat the torpedoes gently, so to speak; that is, fire them with reduced air-flask pressure, at short ranges and at reduced speed, from a boat lying still in the water. This relieves a torpedo of shocks and strains, and diminishes the chances of losing one, while at the same time it can be ascertained whether all parts of the mechanism are in working order. The other way of conducting practice is to charge the torpedo to full flask pressure, adjust it to run at maximum speed, at maximum range, and fire it from a boat running at full speed, at a moving target. The latter way approaches as

nearly as possible to war conditions. It is more severe on the torpedoes, and increases the risk of loss of them occasionally. The great advantage of the more rigorous practice is the training it gives the officers and men. The more such practice we have in peace times, the greater the chances of making successful attacks in war.

The percentage of gunner's mates in the Navy who are expert with torpedoes is small. Therefore, when a boat goes into commission with a green crew, it would be well to start out easy with the torpedoes, and then to work up as quickly as possible to efficient practice. When efficiency has once been attained, every shot should be considered a war shot; that is to say, when the commanding officer presses the key of the electric firing apparatus the torpedo should leave the tube and run straight for the target. In making a real attack the torpedo boat would probably be under fire; there would be no time then to rectify slight errors such as putting in a fresh primer, making a new electric connection, replacing a worn-out battery, overhauling the torpedo-tube mount, or correcting some neglected adjustment in the torpedo itself. Therefore all these details should be carefully looked out for before the torpedo boat reaches the practice ground. There is little to be said in regard to the details of care and handling of torpedoes which is not contained in the books on that subject prepared at the torpedo station and issued to all torpedo boats. These books are probably the most complete and thorough works of this nature in existence, and all persons working with torpedoes should consult them frequently and follow the instructions carefully.

Air compressors are jacked over daily, and are run every time there is steam on a boat. Above all, it should be insisted upon that each boat should use her own compressor to charge her own torpedoes. If this rule is not carried out it will be only a question of time before every compressor in the flotilla will require overhauling. There is also a publication by the torpedo station containing descriptions of, and directions for, running all types of air compressors.

ELECTRICAL OUTFIT.

Most of the torpedo boats are fitted with a dynamo, which is used for lighting the entire boat, running electric fans and supplying current for a searchlight. These dynamos perform their work splendidly, and breakdowns are very rare.

One electrician has been assigned to the flotilla, and, except for jacking the engine over daily (which work is done by the man responsible for the compartment in which the dynamo is installed), he has sole charge of all electrical appliances. Whenever a boat goes out for a trial he goes on her and runs the dynamo during the entire day. Every light and connection is examined, and any defect noted is repaired. After dark of the same day the searchlight is operated, and the running lights are put in place and turned on for a few seconds. Working on only one boat at a time in this way, it would be a very easy matter for one good electrician to keep everything in perfect order.

MANNER OF TRAINING JUNIOR OFFICERS FOR THIS DUTY.

Upon being assigned to duty in connection with the engineer department of torpedo boats in reserve, junior officers should carefully review their engineering text-books. The naval regulations also contain some very valuable information in regard to the care and preservation of machinery. Particularly, however, should all juniors spend as much time as possible in the engine and fire rooms. As the working suit should be the best uniform when any work is to be done, those who have recognized this fact will never hesitate to enter boilers, bilges or crank pits. There should be no reluctance in asking questions of experienced water tenders, coppersmiths, boilermakers and machinists. It is almost an invariable rule that a well-posted practical man in answering questions will give reasons for his opinion, and thus more practical knowledge can be secured from conferences and consultations with skilled artificers than from the study of the ordinary text-book.

LITERATURE OF THE DEPARTMENT SHOULD BE STUDIED.

From time to time Departmental circulars are issued for the guidance of the officers connected with the various torpedo-boat

flotillas. These circulars are prepared with exceeding care, and represent the study, observation and experience of expert officers. The publications of the Bureau of Intelligence, as well as the reports of the bureau chiefs likewise contain valuable information upon service topics. It would also be well for all junior officers to ally themselves with the two semi-official associations within the naval service that have been organized for the purpose of promoting naval efficiency by publishing papers on timely military and engineering subjects. It is often the case that a careful study of a single contribution to a service professional journal will suggest to officers methods of conducting experiments or research that will eventually accrue to their prestige and professional influence.

SOME EXPERIENCES WITH THE U. S. TORPEDO BOAT *GOLDSBOROUGH.*

By Lieutenant H. G. Leopold, U. S. N., Member.

The *Goldsborough* was one of the three torpedo boats authorized by the Act of March 3, 1897. On the 4th of March, 1898, sixteen destroyers and twelve torpedo boats were authorized by the Congress. The construction of these thirty-one vessels subjected their different builders to a collective loss of probably over $2,000,000, three firms having become insolvent as a result of contracting for these boats. While the builders have suffered loss, the Navy Department has actually become the gainer, since it has caused the various shipbuilding firms to understand thoroughly that the Government is not inclined to deal leniently with careless bidders, and that those seeking to undertake naval construction must be prepared to give the Department efficient weapons of war, whether or not it means heavy financial loss to the contractors.

In reference to these boats, Admiral Melville made the following statement before the House Naval Committee:

"About a year previous to the Spanish-American war, not only the Navy Department, but all the shipbuilders conceived the notion that there was a great deal of money in the building of torpedo boats. About this time the several naval powers commenced to demand an increase in speed.

"The Board on Construction in 1897 advertised for three torpedo-boat destroyers, and set a price which the Government would pay. There was a very keen competition for these boats. The several bidders presented their claims in such a manner that the Secretary of the Navy found it difficult to award a contract. In order to secure the contracts, the several builders offered to give more than the Government demanded. A decision was

finally reached to award the boats to the Gas Engine and Power Company at Morris Heights, New York; Wolff & Zwicker, Portland, Oregon, and the Harlan & Hollingsworth Company, Wilmington, Del. As the bids of these firms were very little less than those submitted by the Union Iron Works and the Cramps, of Philadelphia, the latter firms were very much put out that they did not get the contracts.

"Then came the big batch of boats which were contracted for in 1898. There were a great many bidders for these boats, and they scrambled over each other in the effort to get the contracts. Some firms which bid for seven received a contract for three, and they had a grievance because they were not awarded more, when they were the lowest bidders.

"As there had been a bitter fight the previous year for the torpedo boats at the price set by the Navy Department, the Board used these figures as a basis, taking into account the increased cost of material and labor.

"The contracts had hardly been awarded before there was a phenomenal rise in structural material for hull and machinery. Some of the builders, believing that the rise was only temporary, and that by waiting the cost of material would lower, neglected to place contracts for forgings and steel castings. Instead of falling, the cost of material increased. It reached a point in the case of the propeller shafting and machinery forgings that the cost in some cases increased several hundred per cent.

"There was also a progressive increase in the cost of labor, for not only did the wages of the mechanics increase, but the divisions that existed in the several rating classes of mechanics was almost obliterated. Indifferent mechanics demanded to be classed as first-class artisans. These men were not only raised in classification, but the wages of the several classes rose so that in some instances the pay of the poorer classes of mechanics was almost doubled.

"Not only did the cost of material and the price of labor go up, but the several contractors suffered heavy loss by the failure to receive material on time. The mills gave preference to structural shapes for bridges and buildings, for there was more money

in those orders. The indirect losses suffered by the torpedo-boat contractors were very great.

"Such firms as Neafie & Levy, of Philadelphia, and Harlan & Hollingsworth, of Wilmington, Del., companies which had been in business for nearly a hundred years, and which had worked along such conservative lines that they would not take a contract which involved risk, lost tens if not hundreds of thousands of dollars in building these boats.

"For fifteen years the Board on Construction has been preparing plans and specifications for battleships. Probably over $250,000,000 worth of contracts have been given out for warships of various descriptions, and the only people who lost money were the unfortunate torpedo-boat builders who took these contracts. The conditions were abnormal, and no one could anticipate such contingencies.

"The contractors made an appeal to the Secretary of the Navy for relief. This request was referred to the Board on Construction. The Board recommended that the speed requirement be reduced.

"The Board on Construction also recommended that a special Board should be appointed which should investigate the cost of these boats. This Board was presided over by Admiral Ramsay, and submitted a report, which was referred by the Secretary of the Navy to the Chief Constructor and the Engineer-in-Chief. It was found that Admiral Ramsay and his colleagues had gone very carefully into the question, and that they had fixed upon a very fair amount as an average proper cost of the torpedo boats and torpedo-boat destroyers."

Admiral Melville's statement referred to the financial and the structural troubles of the contractors in building these boats, but their difficulties did not end there. In some cases the builders attempted to carry on the preliminary and official trials with crews that were, to say the least, ill-trained to such duties, and as a result the Government secured much valuable information, and naval officers acquired considerable useful experience at the expense of the contracting firms. It is a conservative statement that few of the contractors who undertook to build 30-knot torpedo-

boat destroyers had any realization of the task which they undertook, and, therefore, it is not surprising that impairments, wrecked engines, and even loss of life, resulted from incompetent and inexperienced men attempting to operate the fast-working machinery of the torpedo boats.

EXPERIENCE AND SKILL REQUISITE TO CONDUCT OFFICIAL TRIALS OF TORPEDO BOATS AND DESTROYERS.

Without considering the question of either the design, workmanship or the material used in the construction of fast-working torpedo-boat machinery, it is requisite that a skilled and experienced complement should be secured for running both the builders' and the Government trials. The importance of selecting competent and resourceful men for this work has surely not been realized heretofore in its fullness, and probably there is no better way of impressing builders than by telling of the experiences of the *Goldsborough* while attempting to secure the maximum speed guaranteed by the contractors.

In the collection of data and information necessary for the writing of this article, the writer has corresponded with and interviewed many persons, and has had the free and uninterrupted use of private note books. While some of the information is dependent to a certain extent upon the memory and the general impression of individuals, it is believed that in general the facts are as stated, and although some of the details may not be strictly accurate, such discrepancies do not affect the general record of the *Goldsborough's* various trials.

The history of the *Goldsborough* is not officially complete because the receiver of the Wolff & Zwicker Iron Works, the company which secured the contract, was practically unable to supply any information relating to the unofficial trial trips, due to the fact that neither a log nor a record of the runs was kept. Whatever data was secured was taken by individuals for their own information.

The official history of these boats from the time steam was turned on the engines ought to afford valuable information to naval officers. The writer has learned much from being asso-

ciated with these trials. The careful study and analysis of the various impairments and repeated breakdowns will prove of interest and benefit to all concerned in the design, inspection and the operation of the machinery of torpedo boats.

BRIEF SUMMARY OF VARIOUS TRIALS AT THE DOCK AND IN SHALLOW WATER.

The first preliminary trial was made January 1, 1900, when secured to the wharf, the primary purpose being to clear out bearings and to note the general working of the machinery. As the engines were only run at a speed of 60 revolutions per minute, there was no strain on the machinery, and everything apparently worked well.

At the second trial, this time with the boat under way, the revolutions were gradually increased until the engines were run for a few minutes at about 120 revolutions, when both the H.P. and I.P. crosshead brasses on both engines warmed up, the high-pressure crosshead brasses, particularly, giving trouble. All these brasses continued to be a source of annoyance until they were finally replaced with others, about a year afterwards, at the Puget Sound Naval Station. On this trip it was found that sufficient allowance had not been made for the slip of the link blocks. As a result the links struck the reversing shaft. The connecting links of the valve rods also struck some of the studs of the stuffing boxes; these defects were remedied by cutting off the studs, omitting the jam nuts, and by cutting off one side of the distance pieces of the links. A washer was placed in each end of the reversing engine cylinder to shorten the throw of the links.

A picked crew was secured for the next trial, and the best procurable Cardiff coal was used, and even then steam could not be maintained at a proper working pressure. The vibration of the engines was excessive, although the revolutions did not exceed 260 per minute.

The next trial was made on the Columbia River, a mixture of Cardiff and Cannel coal being used. It is unnecessary to state that such a mixture of fuel did not prove satisfactory, and, therefore, a high pressure of steam could not be maintained. Then

Cannel coal alone was used, with no better results. Steam would be bottled up to about 250 pounds pressure, but there would be 100 pounds fall in pressure when the engines were run at high speed for three minutes or longer. When using Cannel coal the paint was burned off the outside casing of the smokepipe, and a stream of flame would issue from the stack, showing that combustion was completed between the base and the top of the pipe. It seems incredible that anyone who has had experience at sea would attempt to force marine boilers by burning Cannel coal, as this fuel cakes so rapidly that it is practically impossible to force air efficiently through the fires. This experience also taught the novitiates who were conducting the trial the fact that the contract speed guaranteed could never be secured in the shallow water of the Columbia River.

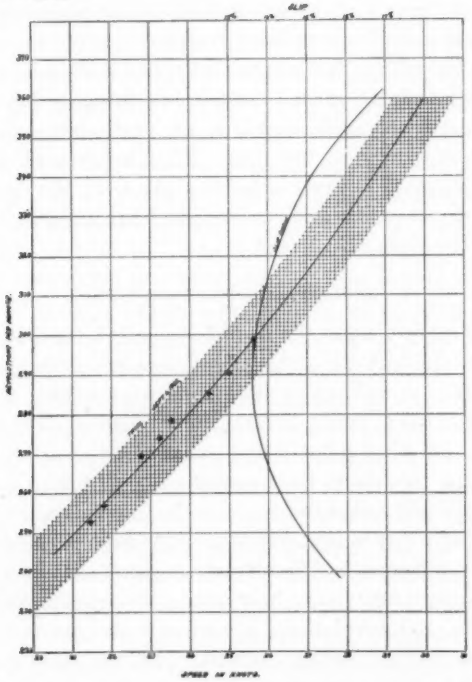
EARLY TRIALS IN PUGET SOUND.

On February 5, 1900, the *Goldsborough* left Portland for Puget Sound, reaching Astoria that night, remaining there some days on account of rough sea on the bar, finally reaching the Bremerton naval station, Puget Sound, February 13. The vessel was docked, cleaned and painted, and an examination of her propellers showed that the blades were considerably distorted.

The standardization trials were begun February 21, 1900, when a single run of one mile was made each way over the trial course. Considerable difficulty was experienced in keeping up steam during these two short runs. When the engines were speeded to 250 revolutions per minute the main valves gave trouble by over-traveling. Both the H.P. and the I.P. crosshead pins heated. The vibration of the engines was excessive, especially in the after portions. The first run over the mile course was made in 2 minutes 13 seconds, the engine speed averaging 302.64 revolutions; the return run was made in 2 minutes and 22 seconds, the revolutions averaging 293.64.

It was attempted to continue the trials on February 22. Before the boat reached the trial course the steam pipe leading to the steering engine gave trouble—where this pipe was brazed to the auxiliary line there was a loosening of the joint. The escaping

steam drove the engine-room force on deck, thereby preventing the shifting from the steam to the hand-steering gear, and the boat was only prevented from running on shore by letting go the anchor. Two months afterwards improvements were effected in the steering gear.



U. S. S. "GOLDSBOROUGH," TORPEDO-BOAT DESTROYER.
REVOLUTION-SPEED CURVE.

PLATE A.

On February 24, 1900, an effort was again made to run the standardization trials. While the engines were running at about 250 revolutions and with a boiler pressure of 250 pounds, the rocker arm on the starboard engine was broken. The valves on both engines, previous to the breaking of the rock-shaft arm,

were striking badly. By reason of the excessive lateral vibration of the engines, combined with other vibrations of the hull and the engine foundations, there was a loosening of the auxiliary steam pipe. The engine vibrations were especially excessive near the after portions. On this trial Pocahontas coal was used, and less difficulty was experienced in keeping up steam.

The *Goldsborough* was then laid up for repairs, the next standardization trials occurring about fourteen months afterwards, in April, 1901. The maximum number of revolutions secured at any time during these latter trials were 343.7, which gave a speed of about 30.84 knots.

On April 30, 1901, the attempt was made to secure an endurance trial of two hours, but fifteen minutes after the start the steam pipe leading to the steering engine broke. After this accident to the steering engine a tripping device was fitted on deck for quickly changing the steering gear from steam to hand power.

Another attempt was made to secure an endurance trial on May 10, 1901, but the engines only ran about fourteen minutes before the L.P. go-ahead eccentric strap as well as the link block broke. Another trial was made May 18th, when the starboard L.P. eccentric rod broke. Prior to this accident the valves were striking badly, although they had been lightened. The engines were run for 1 hour and 25 minutes on June 12th, when the port L.P.-valve rock shaft broke—an examination of the shaft showed old cracks. On this trial the engines were run at the start at a high rate of speed, but the steam pressure soon dropped, and remained low most of the time. There was considerable transverse vibration of the engines, and the low-pressure valves still overrode and struck and were noisy. The crosshead pins heated. It was noticed that No. 3 main-bearing bridge on starboard engine rocked somewhat. The boat was then brought to the navy yard for repairs.

ALTERATIONS AND REPAIRS EFFECTED AT THE BREMERTON NAVY YARD.

The writer of this article was in charge of the steam engineering department of the Bremerton Navy Yard in June, 1901,

when the *Goldsborough* was brought there for a general survey, and all alterations and changes made since that time to the machinery of the vessel have been done under his personal supervision. All trials made subsequent to that time have likewise been personally observed by him.

The Board of Survey appointed by the commandant of the station not only made a careful inspection of everything connected with the boat, but made diligent effort to seek as much information as possible concerning the construction, installation and operation of the main engines, auxiliaries and everything pertaining to the boat. At this period of the *Goldsborough's* history the firm which constructed her were in the hands of a receiver, and therefore there was difficulty in getting her completed. The contractors were still responsible for producing a thirty-knot torpedo boat which could run for two successive hours at that speed. Under such circumstances there were legal and financial difficulties preventing the adoption of radical alterations. Only such changes could therefore be effected which would prevent absolute breakdowns, or which involved the safety of a trial crew.

The principal changes made were as follows:

(1) Renewed both L.P.-valve rocker arms using mild steel. These arms were increased in strength by giving them a parabolic section. The valve rock shaft was likewise increased in diameter to the largest size possible.

(2) The eccentric rods of the low-pressure valve gear were replaced by others of mild steel. Their form was likewise changed to give greater strength.

(3) The crosshead brasses were renewed, the top brasses being made of phosphor-bronze; the lower ones were made of manganese-bronze with Babbit inserted in staggered holes drilled in the composition. All the brasses were increased in strength.

It may be noted that the old brasses were found sprung, which caused them to nip the pin, and thus heat the journals. One of the brasses which was cracked was broken up for examination, and was found quite porous, oil having worked its way almost entirely through the metal. These brasses were subjected to

severe bending stresses, due to the fact that distance pieces had not been fitted between them, and their heating had undoubtedly been due in great part to this cause.

(4) Oil cups were fitted to the crossheads in such manner as to insure efficient and reliable lubrication.

(5) There were installed on the deck some oil tanks whereby the blower engines could be flooded with oil.

(6) The blower fans were fitted with temporary casings. This was done to prevent part of the air discharge from causing light dust to be blown into the eyes of the firemen. Later this casing had to be removed, because it seriously interfered with securing the proper air pressures in the fire rooms when under forced draft.

(7) The starboard and the port engines were secured together by diagonal cross ties, this stiffening being done to reduce vibration.

After these alterations were made no further difficulties were experienced as regards any of the running parts wherein the design had been changed. A reasonable hope was then entertained that a successful trial could be made without any further breakdown.

TRIALS RESUMED WITH A NAVAL CREW.

An official trial was made October 5, 1901, with a mixed crew in the fire room of navy-yard workmen and of volunteers from the U. S. S. *Oregon*. While running down to the course at a fair rate of speed the port L.P. eccentric-strap bolts broke. As the engine was stopped promptly the only other damage done was the bending of the eccentric rod. An examination of the link block showed that the cause of the accident could be laid to the nipping of the link blocks with the link, both the link and the block being of steel. Not only these links, but all the other links of both engines were scored. The blocks were eased off and lined with Babbit metal. A careful examination of all eccentric straps showed that they were weak, and had slightly closed in, although no heating had occurred, and these straps were accordingly replaced by new ones of improved design.

This trial was run with a mixture of Roslyn coal and foundry coke, which, at the beginning, gave good promise of obtaining

fair results. In general, however, mixtures of coal are not likely to prove satisfactory, since every kind of coal requires a special form of firing, and it will be the rare exception when good results can be secured from any mixture of different fuels.

Repairs having been made, the next trial was attempted November 13, 1901. The boat was kept under forced draft for about two hours, and the average revolutions secured were about 304, which, from the standardization curves, gives a speed of 28.1 knots. The main engines worked well on this run. The evaporator, however, vibrated violently as it has done during all the runs that have been observed by the writer. Trouble was experienced in keeping a high steam pressure, although the fuel used on the trip was hand-picked coal fresh from the Franklin mine, Washington.

The next trial was attempted November 15. The boilers were kept under forced draft for about two hours. The maximum revolutions secured were 300. The trial was unsuccessful on account of the fog and rain, and by reason of its being impossible to maintain sufficient revolutions required to make the contract speed. On returning to the Navy Yard after this run the boat rammed the Ordnance wharf, causing considerable damage to the plating of the stem and bow. This accident was probably due to the pilot's lack of experience in handling torpedo boats, thus causing him to misjudge the turning power of the rudder upon encountering an unexpected current. The usual jumping trouble was experienced with the evaporator. The main engines worked well, there being no heating of the bearings.

While running at a speed of 275 revolutions during this trial a loud slapping noise against the hull was heard in the neighborhood of the starboard propeller, although no obstruction appeared to be near that propeller. This noise suddenly ceased on the starboard side, but was taken up by the port propeller with such violence that it was deemed wise to stop the engines and make an examination of both propellers. This inspection showed that neither screw had encountered any obstruction. It was believed that the slapping noise was either caused by water thrown off from the propellers, or by water thrown from one blade inter-

fering with the water of the following blade. When the engines were again speeded the same slapping noise was again heard changing from one side to the other, but when the engines were speeded to about 300 revolutions this noise disappeared. As this unusual noise has been heard on all trials, its cause may be definitely accounted for by those who have made a special study of the question of the screw propeller. It may be incidentally stated in connection with this subject that both propeller shafts fitted their stern bearings and struts neatly, and that there was no undue play.

Another attempt to secure a trial was made September 10, 1902. While the machinery was being speeded preparatory to the trial, the steam pipe leading to the evaporator worked loose at its junction with the auxiliary steam pipe. The cause of this action was due to vibration of the evaporator, and as a result the trial had to be abandoned. A further attempt was made to secure a trial on September 12, but the effort had to be abandoned on account of the accumulation in the atmosphere of smoke from forest fires.

WRECKING OF THE PORT ENGINE.

Four days later, September 16, the last trial made with the *Goldsborough* was attempted. On this trial the propellers were standardized to the speed shown on the standardization curve. While making the last run over the measured mile it was impossible to determine the speed secured by reason of the mist which obscured the range signals. On this run the engines made about 315 revolutions. By reason of the condition of the weather it was determined to run the hour endurance trial, and to complete the standardization tests when conditions were favorable as to weather, state of sea and visibility of the range signals. The boat was then steamed on the course, the engine speed being set for 300 revolutions. About two-and-a-half minutes after the endurance trial began, but even before the engines had attained the desired speed, the port low-pressure valves began to knock. The valves and the pistons suddenly began to pound, followed by a violent shaking of the engines which was of such nature

as to throw the lagging off the main steam pipes. The throttles were rapidly closed. The complete wrecking of the port engine, however, occurred, but the casualty was so sudden and unexpected that those on duty in the engine room were unable to determine which part had broken first.

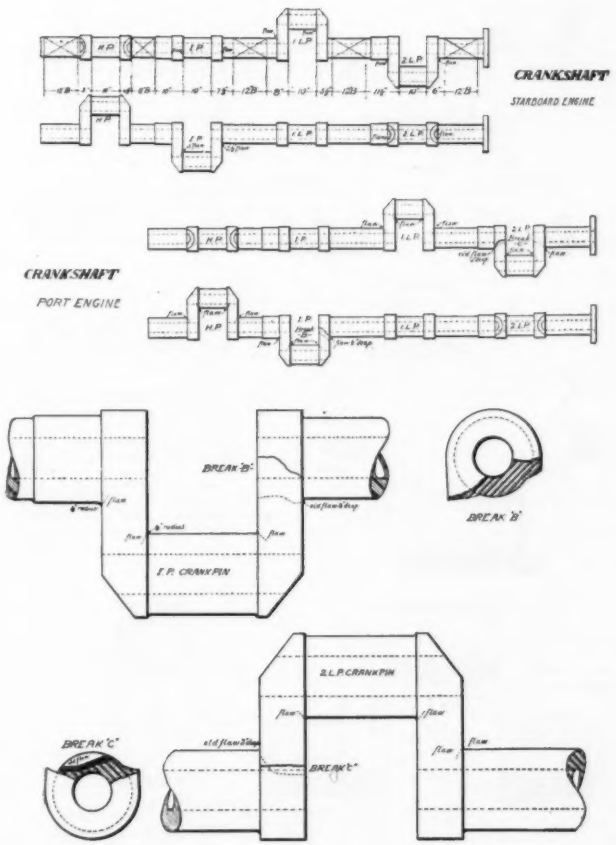
EXTENT OF THE WRECKAGE OF THE PORT ENGINE.

Upon examining the wrecked engine the following impairments and casualties were found to have occurred:

1. The forward web of the after low-pressure crank was broken, and it was observed that the two parts of the web had rode over each other for some revolutions and until the engine was stopped. This readily accounts for other casualties which were probably a sequence of the breaking of the web of the crank. The order in which the several breaks occurred can only be a matter of conjecture. A careful examination of the fracture of the crank web showed that previous to the accident the web had been partly broken, causing a reduction of at least five square inches in the cross sectional area of material. The photograph of the break plainly shows that there was a crack in the rough forging, and that this crack was progressively extended on the successive trials where the engines had been run at a high speed. The successive extensions of the crack as a result of the several trials can be clearly outlined, the appearance of the metal changing as each extension occurred.

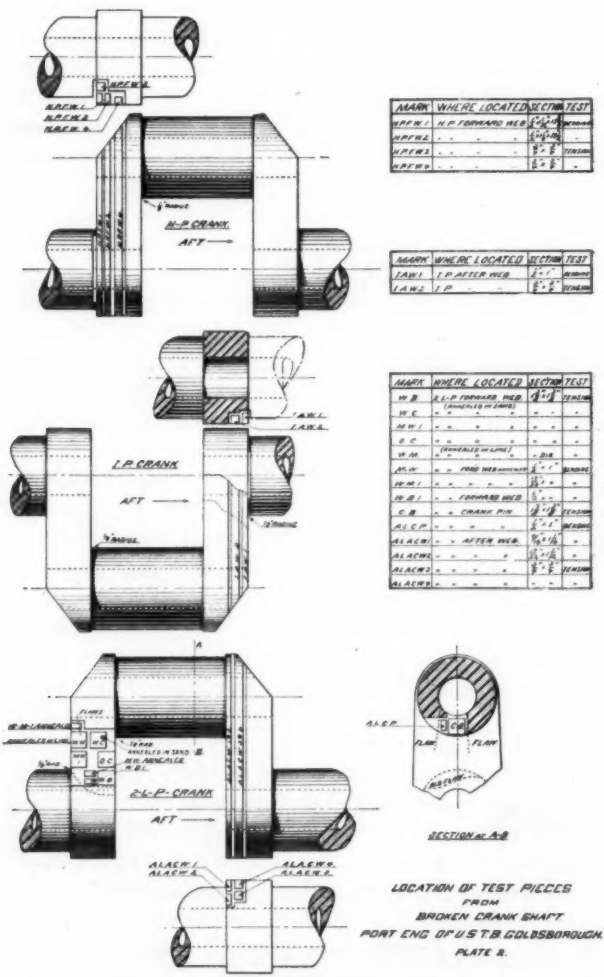
Referring to Plate 1, it will be observed that the flaws of the port L.P. and I.P. shaft are located where we should expect to find them with the engine keelsons yielding and the main bearing bridges rocking with the keelson. It will also be noted that the break "C" of the port L.P. shaft is about where we should expect it to occur considering the location of the evaporator that suddenly checked the heavy vibrations of the engines.

Plate No. 2 shows the location of various test pieces taken from the broken shaft. The results of these tests are given in the accompanying Table No. 1 and are self-explanatory, clearly showing the difference in the composition of the metal due to the treatment it received and the strain it underwent. The com-



FLAWS & BREAKS IN CRANK SHAFTS
U.S.S. GOLDSBOROUGH

PLATE 1.



U. S. S. "GOLDSBOROUGH," TORPEDO-BOAT DESTROYER. LOCATION OF TEST PIECES.

PLATE 2.

TABLE I.—SUMMARY OF TESTS OF PIECES TAKEN FROM PORT CRANK SHAFT OF UNITED STATES TORPEDO-BOAT *GOLDSBOROUGH*, NAVY YARD, PUGET SOUND, WASHINGTON, NOVEMBER, 1902.

TABLE I.—SUMMARY OF TESTS OF PIECES TAKEN FROM PORT CRANK SHAFT OF UNITED STATES TORPEDO-BOAT *GOLDSBOROUGH*, NAVY YARD, PUGET SOUND, WASHINGTON, NOVEMBER, 1902.

TENSILE TESTS.

No.	Mark on piece.	Length over all.	Shape of ends.	Diameter.		Area.		Length between gauge marks.	Actual load.		Load, per square inch.		Elongation.	Reduction of area.	Time.	Taken from.
				Before test.	After rupture.	Before test.	After rupture.	Before test.	Elastic limit.	Maximum load.	Elastic limit.	Maximum load.				
		ins.	*ins.*	*in.*	*in.*	*in.*	*in.*	*ins.*	*lbs.*	*lbs.*	*lbs.*	*lbs.*	*p. c.*	*p. c.*	*sec.*	
1	W. B.	8½	1⅜×1⅜	.5	.475	.19635	.17721	2	2.21	19,800	19,890	62,136	101,300	10.5	9.75	65	For'd web A. L. P. crank.
2	M. W. I.	8½	1⅜×1⅜	.4995	.4922	.19596	.19027	2	2.06	12,250	15,265	62,520	77,900	3	2.94	48	For'd web A. L. P. crank.
3	W. C.	8½	1⅜×1⅜	.4993	.4908	.1958	.18919	2	2.1	10,100	13,630	51,580	69,600	5	3.38	42	For'd web A. L. P. crank.*
4	W. M.	8½	1⅜ dia.	.5	.495	.19635	.19244	2	2.04	8,535	8,535	43,470	43,470	2	1.99	50	For'd web A. L. P. crank.†
5	O. C.	8½	1⅜×1⅜	.5	.4905	.19635	.18896	2	2.06	11,000	16,535	56,027	84,200	3	3.76	47	For'd web A. L. P. crank.
6	C. B.	10½	1⅜×1⅜	.5	.37	.19635	.10752	2	2.47	12,100	19,865	61,623	101,180	23.5	44.99	95	A. L. P. crank pin.
7	A. L. A. C. W. 3.	8½	⅜×⅜	.4995	.398	.19596	.12441	2	2.36	14,000	21,625	71,420	110,360	18	36 51	65	Aft. web A. L. P. crank.
8	A. L. A. C. W. 4.	9⅛	⅜×⅜	.5	.374	.19635	.10986	2	2.42	14,200	20,610	72,320	104,970	21	44.05	110	Aft. web A. L. P crank.
9	I. A. W. 2.	9⅛	⅜×⅜	.499	.385	.19556	.11642	2	2.4	13,800	21,310	70,563	108,980	20	40.47	107	Aft. web I. P. crank.
10	H. P. F. W. 3.	12⅛	⅜×⅜	.4995	.382	.19596	.11461	2	2.4	14,100	21,120	71,960	107,800	20	41.52	92	For'd web H. P. crank.
11	H. P. F. W. 4.	11⅛	⅜×⅜	.4993	.38	.1958	.11341	2	2.39	13,700	21,080	69,962	107,670	19.5	42.08	90	For'd web. H. P. crank.
				Mang anese		bronze	test pie ce cut f rom cra nk-shaf t beari ng No.					4, port		engi ne.			
12		9½	⅜ dia.	.498	.457	.19478	.16403	2	2.35	7,980	11,730	40,970	60,230	17.5	15.79	97	Main bearing, No. 4.

BENDING TESTS.

No.	Mark on piece.	Section.	Length.	Number of breaks.	Fracture between sections numbered.								Test piece taken from.
					1 and 2.		2 and 3.		3 and 4.		4 and 5.		
					Angle.	Diam.	Angle.	Diam.	Angle.	Diam.	Angle.	Diam.	
		ins.	*ins.*		*degs.*	*in.*	*degs.*	*in.*	*degs.*	*in.*	*degs.*	*in.*	
1	W. B. I.	1 ×⅜	8½	1	Broke s	hort off i	n jaws of	vise.	...	...	...	...	For'd web A. L. P. crank.
2	M. W.	1 ×⅜	8½	3	Broke s	hort off.	28	...	14	...	...	...	For'd web A. L. P. crank ‡
3	W. M. I.	1 ×⅜	8½	4	20	...	18	...	10	...	30	...	For'd web A. L. P. crank.‡‡
4	H. P. F. W. 1.	1 ×⅜	13½	2	175	1	Nicked.	Broke sh	ort off.	...	...	...	For'd web H. P. crank.
5	H. P. F. W. 2.	1 ×⅜	13½	1	180	1	...	...	...	...	...	...	For'd web H. P. crank.
6	A. L. C. P.	1 ×⅜	11½	3	180	⅜	16 N	icked.	110	...	...	...	A. L. P. crank pin.
7	I. A. W. I.	1 ×⅜	9½	1	180	⅜	...	...	...	...	...	...	After web I. P. crank.
8	Aft low, aft C. W. 1.	1 ×⅜	10	2	180	⅜	Nicked.	Broke sh	ort off.	...	...	...	After web A. L. P. crank.
9	Aft low, aft C. W. 2.	1 ×⅜	11	...	Bent do	uble to a	diameter	of ⅜ inch	without	breaking.	...	...	After web A. L. P. crank.

* Annealed in sand. † Annealed in lime. ‡ Annealed.

position of the various specimens taken from different parts of the shaft would clearly indicate that the shaft had been injured in forging or had become crystallized by the severe strains to which it was subjected during the repeated trial trips. An examination of the various specimens likewise showed that no satisfactory nickel alloy had been formed; that there was simply a mechanical mixture of the elements, which was evidenced by the segregation of the nickel.

Plate No. 7 illustrates the general distorted condition of the port engine, and particularly the broken parts.

The photographs of Plates Nos. 3 and 4 illustrate the broken crank shaft after its removal from the boat.

2. The arm of No. 4 bearing bridge was cracked, this being a development of an extensive flaw that had apparently existed for some time. The accompanying illustration will clearly show the condition of affairs. Plate No. 8 illustrates the extent of this old flaw in No. 4 bearing bridge.

3. The H.P., the I.P. and both L.P. cylinders were torn apart, as shown in the following photograph. Plate No. 5 illustrates this rending asunder of the two low-pressure cylinders, due to the over-riding of the parts of the broken crank shaft, and to the subsequent breaking of No. 4 main bearing bridge.

Photographic Plate No. 6 illustrates the rending asunder of the H.P. and I.P. cylinders.

4. Nos. 2, 3 and 5 main-bearing bridges were badly sprung and forced down.

5. The after web of the I.P. crank shaft was broken through. An examination of this rupture showed that the crank was a development of a flaw that had existed for some time.

6. The forward L.P. valve was broken.

7. The after low-pressure valves were damaged from having struck the steam-chest cover.

8. The rocker arm carrying the forward low-pressure valve was bent.

9. The angle frames under the engines were warped and drawn together. Plate No. 9 illustrates the abrupt ending of the outboard engine girders on a single floor frame of the hull

FIG. 3.—U. S. S. "GOLDSBOROUGH." BROKEN CRANK SHAFT.

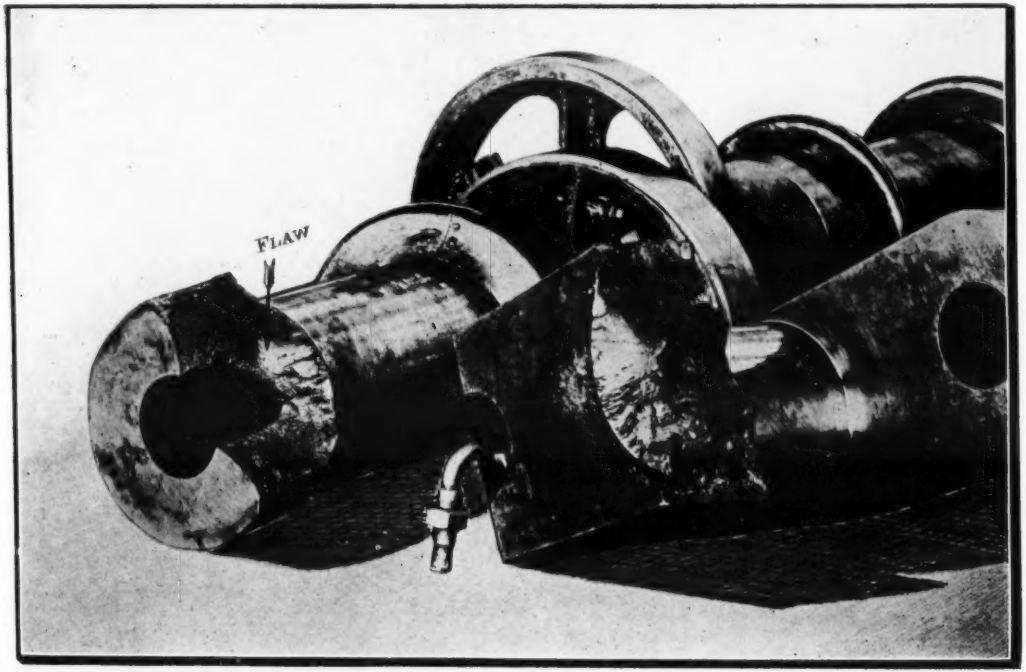


FIG 4.—U. S. S. "GOLDSBOROUGH." SHOWING INDICATION OF OLD FLAW AND FINAL BREAK IN CRANK SHAFT.

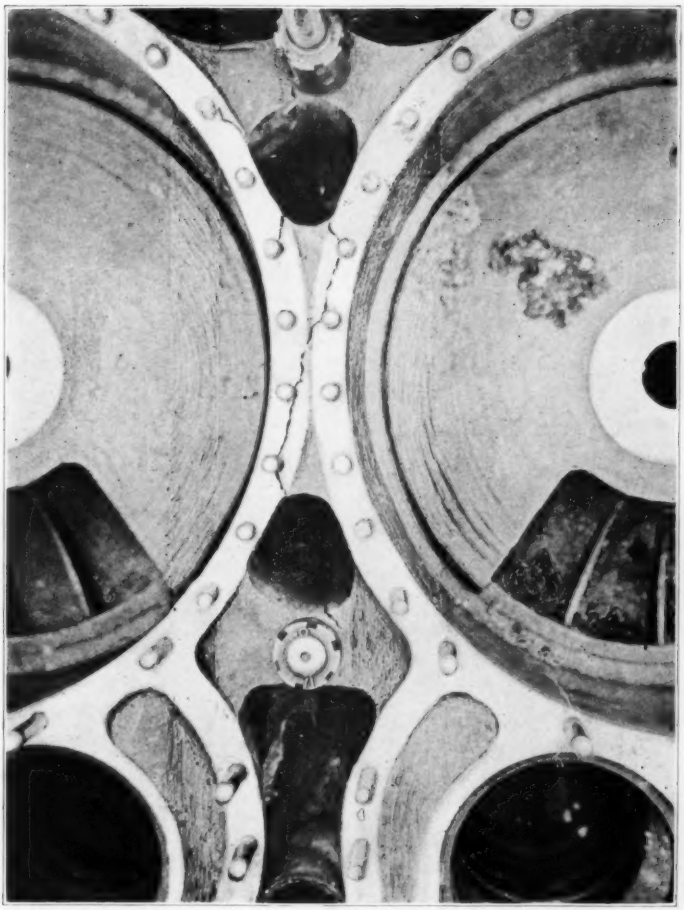


FIG. 5.—U. S. S. "GOLDSBOROUGH." LOOKING DOWN INTO OPEN CYLINDERS. SHOWING CRACKS IN CASTINGS BETWEEN CYLINDERS.

FIG. 4.—U. S. S. "GOLDSBOROUGH." SHOWING INDICATION OF OLD FLAW AND FINAL BREAK IN CENTRAL CASTING.

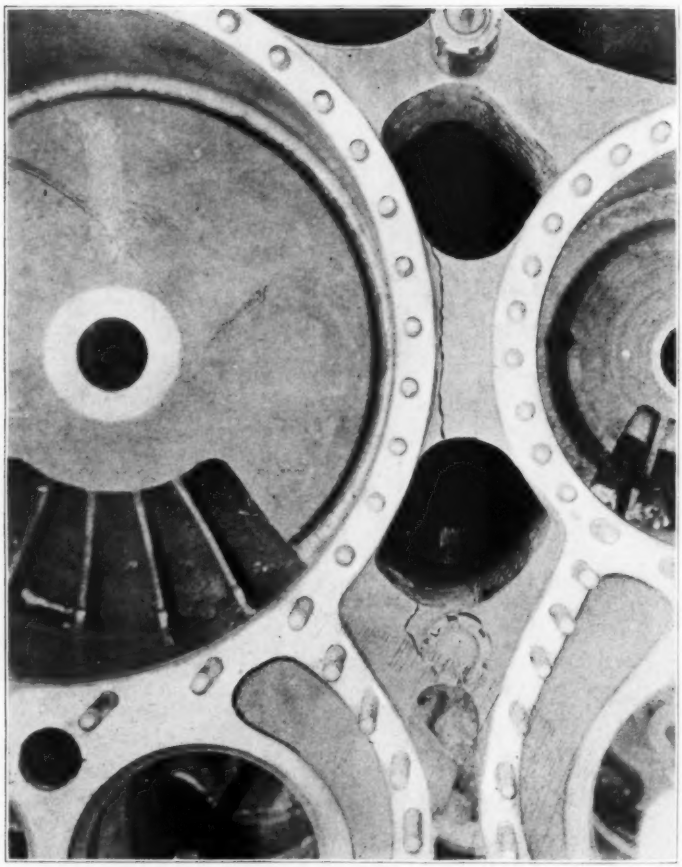


FIG. 6.—U. S. S. "GOLDSBOROUGH." LOOKING DOWN INTO OPEN CYLINDERS. SHOWING CRACKS IN CASTINGS BETWEEN CYLINDERS.

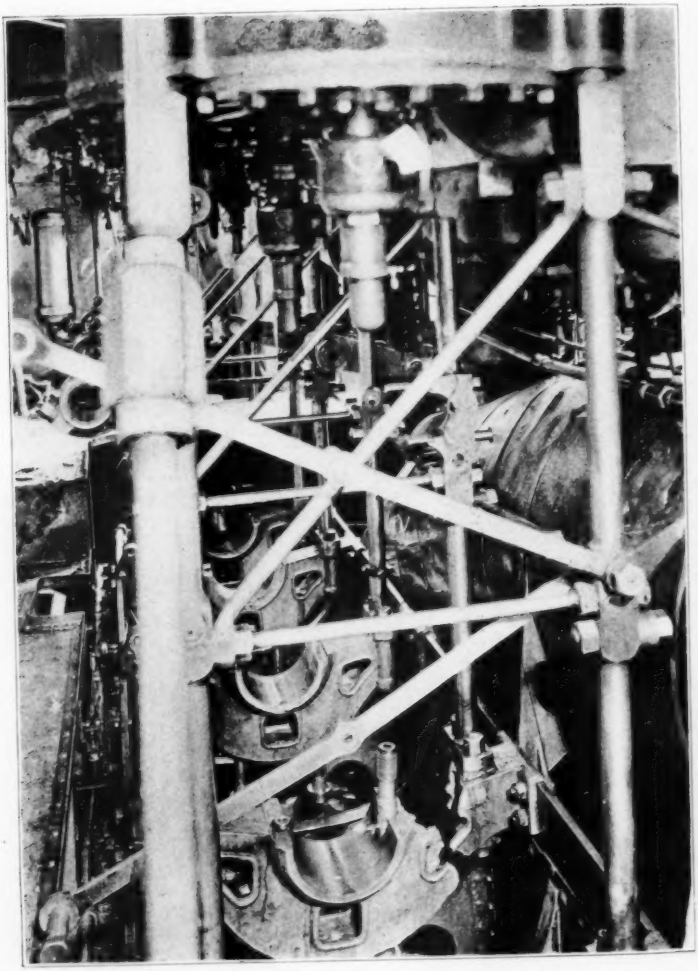


FIG. 7.—U. S. S. "GOLDSBOROUGH." FRACTURE OF PORT AFTER BEARING BRIDGE.

structural system, and which had not been specially supported. It is believed that the arrangement of these girders was responsible in great part for the failure of the vessel to make a satisfactory record.

10. The vertical ties and the engine framing were sprung.

11. The forward L.P. valve stem was found bent.

12. The distance pieces between the forward low-pressure valves were bent,

13. The I.P. and both the L.P. pistons were driven down about three-quarters of an inch on the piston rods. The bosses of all the pistons were cracked.

14. Practically all the fore-and-aft athwartship engine braces were sprung.

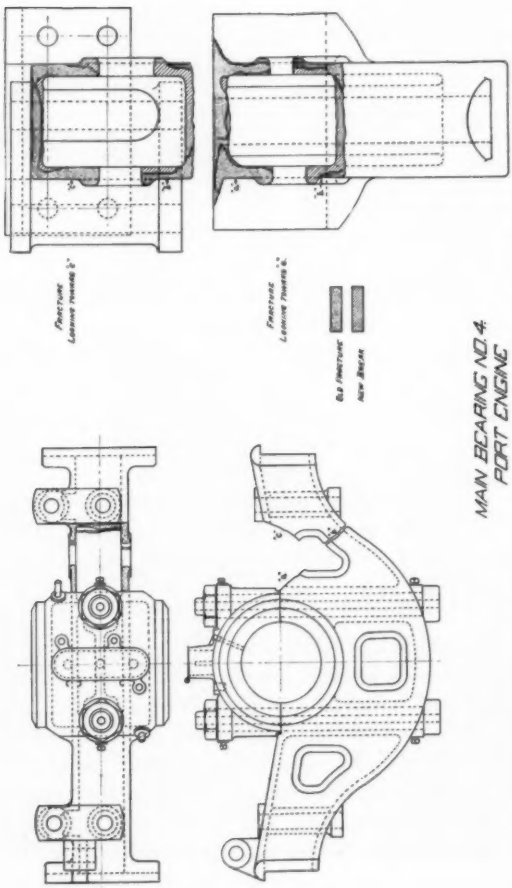
15. The coupling and the main-bearing bolts, as well as the crosshead brasses, were somewhat damaged.

16. It was discovered that the bolts of No. 4 main-bearing bridge had been working previous to the accident, thus permitting the bridge to rock to some extent. Other main bearing bridge bolts were also found to have been working to a slight extent.

17. Cracks of considerable extent were found in the port main-crank shaft. When the starboard engine was stripped, cracks of the same nature were found in the shaft of that engine.

PROBABLE CAUSES OF THE ACCIDENT.

From the testimony of those on duty in the engine room when the accident occurred, and from an examination and study of the wreck, it appears that both the low-pressure crank web and No. 4 main bearing bridge were yielding under the strain to which they were subjected when the engine was running at high speed. As previous defects existed in both these parts, it is not possible to determine positively whether the web or the bridge yielded first to the undue strain to which they had been subjected by reason of being unimpaired. The giving way of either the crank web or the main bearing bridge would have been sufficient to cause the extended wreckage that occurred. The character of the fracture of the crank web, and the consequent over-riding of the



U. S. S. "GOLDSBOROUGH," TORPEDO-BOAT DESTROYER. FRACTURED BEARING BRIDGE.

PLATE 8.

two sections of the broken shaft, appear to furnish the most plausible explanation of the subsequent damage to the other parts.

In order to determine whether the breaking of the shaft was due to defective material in the original billet of the forging, or to the overheating of the forging during manufacture, test pieces were cut from various parts of the shaft. The places from which the test pieces were cut are shown in the accompanying illustration, and the analyses of the various test pieces are given in the table which follows. Four test pieces were taken from the shaft for chemical analysis. The general appearance of these test pieces varied considerably, and, when carefully examined under the magnifying glass, it was clearly manifest that the nickel of the alloy was not uniformly distributed throughout the material.

No.	Sample.	Carbon.	Nickel.	Phosphorus.	Sulphur.
			per cent.		
1	A.L.P. forward web.....	...	4.48	...	.048
2	A.L.P. at fracture	.429	3.75	.039	.030
3	I.P. after web at fracture.	.423	3.07	.083	.038
4	H.P. forward web.....	.524	2.36	.081	.040

Test piece No. 1, a small crystal about the size of a pea, was of a silver-white appearance, which at first glance appeared to be pure nickel that had failed to enter into chemical combination with the steel. It showed a higher percentage of nickel than any other of the samples which have been tested. The fracture upon examination showed small bright spots of a nature similar to this specimen, thus probably establishing the fact that there had been no successful incorporation of the nickel into the steel. The variation in the amounts of carbon, phosphorus and sulphur in the various specimens would also indicate that the material was a mixture rather than an alloy of nickel and steel.

An examination of the fracture of the L.P. crank web, together with the analysis of the test piece of the material, gives almost conclusive proof that this shaft was injured in forging.

As to the other cracks observed, the same explanation cannot apply, but they would more likely be due to the small radius of

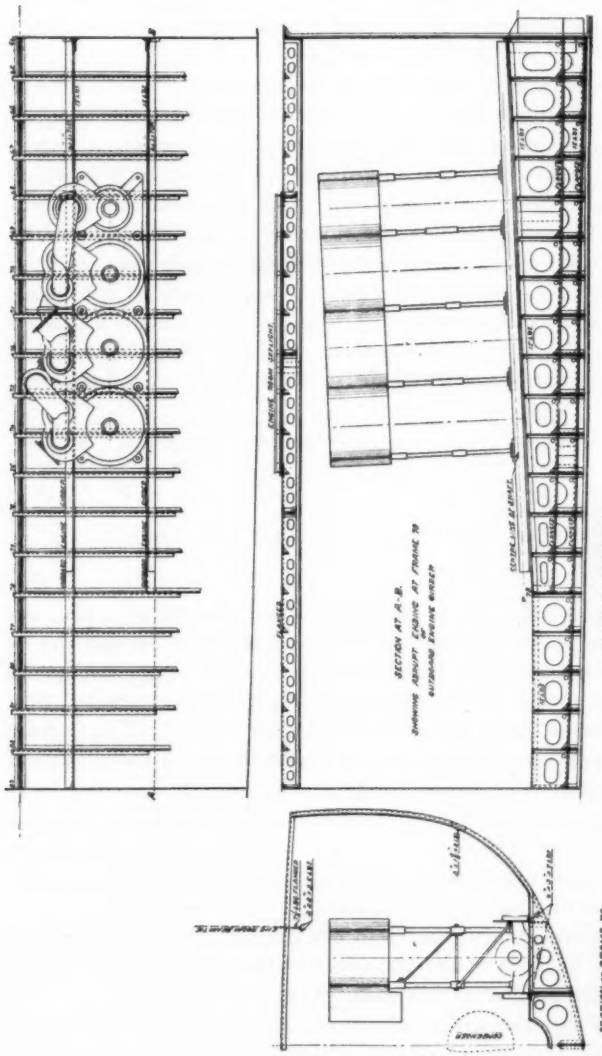


FIG. 9.—U. S. S. "GOLDSBOROUGH" TORPEDO-BOAT DESTROYER. RELATION OF ENGINE FOUNDATIONS TO FLOOR MEMBERS.

the fillets of the journals, in some cases being as small as one-eighth of an inch. The importance of having well rounded fillets for crank shafts in machinery of this character cannot be overestimated.

It is desired to call special attention to the result of breaking up under a steam hammer of one of the crank webs. By making a slight nick this web was readily broken in a manner similar to the fracture which occurred in the low-pressure crank web. There was no appreciable elongation, the fracture resembling more the appearance of white cast iron rather than that of steel. This test was made to secure some idea as to the power of nickel-steel to withstand sudden shock, and it conclusively showed that while test pieces might show, under slow test and strain, a higher tensile strength and give a fair elongation, the material would break short without any appreciable elongation when subjected to sudden shock.

GENERAL CAUSES OF THE FAILURE OF THE MACHINERY OF THE *GOLDSBOROUGH*, AND IMPORTANT FACTORS THAT MUST BE CONSIDERED IN SUCH MACHINERY DESIGN.

By referring to the plan of the engine keelsons, Plate 9, it will be observed that the outer keelson extends from frame No. 64 to frame No. 78, and that it is firmly secured at the forward end to a vertical bulkhead by means of brackets. The after end of the keelson lands on a simple frame which has no special support. On the port side of the engine room is located the evaporator, and this auxiliary is about in line with the direction in which the vibrations of the engines would be thrown into the after part of the hull. On one trial trip the vibrations of the evaporator were so violent as to break loose its steam supply pipe, and on another trial the steam pipe of the evaporator was disconnected, since it was feared that the serious vibrations would result in a casualty. Reflection and experience would suggest that no heavy weights like an evaporator should be located in the engine rooms where they would interfere with any vibrations set up by the engines. The vibrations of the engines should be gradually absorbed by the hull, for it is evident that

if such vibrations were suddenly checked, reverse vibrations would be set up which would cause both the hull and the engines to be subjected to sudden shocks of great violence.

The engine keelsons should be carried as far forward and as far aft as possible, landing at last on a vertical bulkhead that should be sufficiently stiff to carry the engine vibrations well into the hull. Such construction would disseminate the engine vibrations throughout the ship, thus avoiding any point of local weakness. In this respect the *Goldsborough* was particularly faulty, the outboard engine keelson stopping short on a simple, ordinary frame, while a heavy evaporator was located in the port engine room at a place that was approximately the weakest point in the hull.

The low-pressure valves of the main engines were carried by a single eccentric and rocker shaft, and they weighed, originally, with spreaders and links, somewhere between 500 and 600 pounds. The effect of vibrating valves weighing a quarter of a ton, 600 to 680 strokes a minute, through a path of 6¾ inches, is necessarily dangerous in light machinery. The design of these valves alone would account for the greater part of the vibrations, if it does not account for all the difficulties experienced with the low-pressure valve gearing. Furthermore, the weight of these valves was in no way relieved by the employment in the design, of balance valve pistons, made on the continuation or upper end of valve stems. The valve chamber or cylinder formation was faulty, in that the valve cylinder walls were not continuous, the upper and lower ends of valve cylinders having the working surface bored out of a prolongated formation cast on to the engine-cylinder covers and bottoms. That the working surfaces of the valve liners were ever successfully placed in line is a feat in itself, for the tightening down of a single cylinder bolt might have thrown the valve-liner surface out of parallel.

The link blocks of these engines were of steel, and gave trouble by nipping the links, and from this cause the blocks were scored badly. On one occasion the link blocks broke, and this must have brought a very great strain on all parts of the valve gear. After the link blocks were lined with Babbitt metal no further

difficulty was experienced; therefore when the blocks are made of steel the precaution should be taken to line them with some anti-attrition metal.

Where two engines are placed back to back in the middle of a vessel, leaving only two outboard passageways, it would appear advisable to use diagonal tie rods for stiffening the engines, thereby giving the engines practically a common base. By thus stiffening the engines they would vibrate as a unit rather than independently of each other.

The arrangement and location of the auxiliary steam and the exhaust pipes are also matters of importance, since there is a tendency at all times for these pipes to shake considerably. Where auxiliary pipes branch off from the main steam pipe the loosening of some of the joints is ever liable to take place, which may cause a serious enough leakage of steam to compel the shutting off of the entire auxiliary line.

DOUBTFUL VALUE OF NICKEL-STEEL.

In the construction of fast-working machinery of the torpedo-boat type the practicability of using nickel-steel is exceedingly doubtful. The specimens of this steel that were cut from the *Goldsborough* conclusively showed that the metal would not stand shock, although the same metal when struck with innumerable mild blows could be bent to a very short radius. Whether the brittleness exhibited under the bending stress would affect its tensile strength under sudden shock could not be determined. It might be advisable to make a series of experiments showing how this metal would act as compared with mild carbon-steel. In all probability the milder carbon-steel would show its superiority, since it would not yield so readily to shock. Nickel-steel, however, makes a very superior bearing, and, as shown on the testing machine, where it was exposed to a slow pull, it possesses great strength. It is but just to the manufacturers of nickel-steel to state that that which was used on the *Goldsborough* had been manufactured several years, and that since that time improvement has undoubtedly been effected.

MANAGEMENT OF TORPEDO-BOAT MACHINERY.

In the management of torpedo-boat machinery, those in charge should ever keep in mind the excessive power developed per weight of engine, and the weaknesses of hull and of engine foundations. Changes of speed should therefore be made gradually, otherwise undue and dangerous strains will be brought into play which are almost certain to cause impairment or damage. The inexperienced are sometimes likely to bottle up steam and increase the speed suddenly, and the danger of such a procedure ought to be made evident to any one placed in charge of such machinery. The futility of such action should be recognized when one takes into consideration the small amount of steam room in the boilers of torpedo boats. The increase in engine speed should therefore be progressive, and care should be taken before doing so that the fires have been put in such condition that the high speed could be maintained for a considerable time.

Skill, intelligence and experience are essential in the fire room. The water tenders should be resourceful, and should be thoroughly familiar with all the piping, both above and beneath the fire-room floor plates. Uniform firing should be carried on so that the boiler tubes would be exposed as far as practicable to uniform conditions.

After making some of the preliminary runs with the *Goldsborough* it was observed that most of the products of combustion passed between the back half of the tubes, and that a large number of the tubes at the front were covered with soot, while the back tubes showed conclusive evidence of having been exposed to intense heat. The firemen were then instructed to carry heavy fires at the back of the furnace and to keep them backed up against the rear tubes; it was further directed that the fires should taper from the back towards the front. This method of working the fires proved quite satisfactory, since it enabled the firemen to make sufficient steam to run the engines about 250 revolutions under natural draft alone. On firing in this manner when running under forced-draft conditions steam could be readily kept up to the desired maximum pressure, even render-

ing it necessary at times to slow the blowers to prevent the lifting of the safety valves.

When running at maximum speed reliance should not be placed upon wick oil feeds, but all bearings should, as far as possible, be kept flooded. There can be no economy in the consumption of oil when running machinery of this character, for a shortage of oil on any bearing or running part will cause damage to occur in remarkably short time. The whole oiling arrangement should be of such character that those responsible for the lubrication of the engine may be assured that the oil reservoirs are of sufficient capacity, and that the oil pipes are of adequate size to assure a regular supply for a period long enough to permit the oiler to give special consideration to some heated part for at least a few minutes.

LOCALITY AN IMPORTANT FACTOR IN EFFICIENT BUILDING OF TORPEDO-BOAT MACHINERY.

A review of the history of this torpedo boat and of her numerous trial trips will clearly show that there should be a suitable trial course within a reasonable distance of the plant where such machinery is constructed. A general history of the torpedo boats that were authorized by the Congress in 1897 and in 1898 affords conclusive evidence that the cost of the trials was several times greater than was anticipated. Where it takes a day or two to reach the trial course, it is often the case that rough weather is encountered before the boat reaches her station, and thus the builders are subjected to vexatious delay and great expense in waiting for good weather. Then, again, some minor repairs may have to be made upon reaching the course which cannot be done by the small shops in the vicinity, and long delay often results from a series of apparently minor accidents.

THE IMPORTANCE OF SECURING AN EFFICIENT TRIAL CREW.

About seven years ago the Columbian Iron Works, of Baltimore, brought from New York a picked crew of mechanics and firemen for running the trials of the *Foote*, *Rodgers* and *Winslow*. The chief engineer in charge of this crew has run the trials of

nearly a dozen torpedo boats, and there are many who believe that such trials should only be carried on by men who have had his extended experience in such work. It is obvious that when such a crew is employed the men may be under pay for months, due to the fact that so much tuning of the machinery is requisite before the engines can be made ready for the endurance and maximum-speed standardization trials. The cost of running the trials must be regarded by all the contractors as an important item of expense, and this should be taken into consideration in submitting a bid for the construction of such boats.

The mechanics who are sent into the engine rooms, and the firemen who enter the stokeholds, to carry on such trials, must be resourceful in emergency, and thoroughly familiar with the machinery which they are called upon to operate. They must keep their ears sensitive to unusual sounds, their eyes open for leaky joints, and they must also realize that the sense of touch must determine whether there is a tendency of journals to heat. Even the sense of smell must be exercised to note if burning oil is present from excessive friction. From time to time the water should be tasted to find out whether it is salty. All who go in engine rooms or fire rooms are thus subjected to both physical and mental strain, and they well earn the pay that is given them.

CHARACTER OF COAL USED A DETERMINING ELEMENT IN SECURING THE MAXIMUM SPEED OF TORPEDO BOATS.

When attempting to make the maximum standardization trials, or in the effort to run off the endurance test, hand-picked Pocahontas or Cardiff coal should be used. If such coal cannot be procured in the local market, then arrangements should be made to bring several car loads from distant points. The builders should also keep in mind that it will be economy to procure fifty tons too much rather than a ton too little. During the past ten years at least a half dozen endurance trials of torpedo boats have been discontinued because the builders were short two or three tons in the bunkers. It sometimes requires several hours to get engines and boilers in readiness for the maximum endur-

ance trial, and during this period there is a drain upon the hand-picked coal. The minimum expense of conducting a trial trip of a boat of the *Goldsborough* class will approximate $1,000. And the actual cost of the coal is therefore but a small percentage of the total expense involved in carrying on such trials. Some builders attempt to stow both a cheap and a good quality of coal in the bunkers, believing that just before the trial commences it is only necessary to feed the fires with the superior fuel. The exercise of a little common sense ought to show that if the bed of the fire is of inferior fuel that the fires will either clinker or become dirty, and that the chances are remote that the boilers will evaporate the expected amount of water. It has been remarked by one official who has had much to do with the trial of torpedo boats, that if, on the day when the supreme effort is expected of the boilers and of the machinery, there is any poor coal in the bunkers, it will surely be thrown into the furnaces just when it should not be supplied.

GENERAL CONCLUSIONS.

The history of the ill-fated *Goldsborough* should be a forcible object lesson to small shipbuilding firms who have had no experience with naval work, as to the inadvisability of seeking contracts for the building of torpedo boats. On board every type of warship there are limitations as to weight, height, and the floor space allowed the installation of machinery; and unless either the administrative officers, or the leading technical experts of the shipbuilding company, have had experience in the construction of such machinery, it is certain that such firms cannot take the work without running the risk of suffering heavy financial loss.

There are military reasons why the Navy Department should encourage as many firms as possible to engage in naval construction, so that in time of emergency there could be secured great rapidity in the speed of construction of all classes of war vessels. Taking advantage of this desire of the Government to widen the field of naval construction, inexperienced bidders have submitted low proposals, and insisted upon their right to be awarded the

contract. The experience of the builders of the *Goldsborough*, and of other torpedo boats, will make it easier work hereafter for the Navy Department to throw out the proposals of bidders whose ability to build such craft may be doubted.

The Wolff & Zwicker Iron Works, where the *Goldsborough* was built, had been doing a prosperous business previous to taking the contract for this torpedo boat. At times the firm had employed as many as 500 workmen. The taking of this contract practically ruined the firm, for the *Goldsborough* has yet to fulfill contract requirements, and this may yet involve the expenditure of large amounts of money.

Even after those boats that have encountered a series of mishaps meet contract requirements, they are always looked upon as untrustworthy machines. Both officers and men who are detailed to duty with such vessels are apprehensive as to the endurance and safety of the machinery when subjected to severe work.

Every torpedo boat which has required years to tune her up to the point of making a successful trial trip, is in some respects a discredited fighting machine, and the boat will have to perform successful service for several continuous years before any flag or any commanding officer will have confidence enough in her to detail her for work where reliance and endurance are necessary.

One of the most important points that should be kept in mind in designing torpedo boats is that very powerful machinery is to be installed in a hull of extreme lightness. Special care should therefore be taken whereby the strains and the vibrations of the engines may be taken up throughout the entire hull. The interposition of any heavy weights that may tend to interfere with the uniform distribution of the engine strains and vibrations into the hull as a whole should, as far as possible, be avoided. It is certain that if such strains are taken up by comparatively few of the ship's frames that racking of the engines and the hull will follow, and that some serious accident may be anticipated.

The engines should be balanced as far as possible, not only as

regards the pistons and the connecting rods, but in matters affecting the strains and the vibrations of heavy valves.

Only the best quality of material can be expected to prove satisfactory in such construction. The best of steel is liable to injury during the process of forging, and the best of compositions may be ruined in the process of casting, therefore an important duty and responsibility rests upon the inspector of material.

The following analyses of chips from the brasses of the main bearing bridge and the lower crosshead shows the character of the constituents that have entered into their composition:

Chips from Main Bearing Bridge.

Aluminum,	.12
Copper,	56.95
Iron,	1.50
Lead,	.03
Manganese,	.08
Tin,	1.06
Zinc,	40.26
	100.00

Crosshead Brass Chips.

Copper,	85.70
Iron,	.05
Lead,	.10
Tin,	9.91
Zinc,	4.24
	100.00

In conclusion, it may be said that in designing, constructing, installing, and operating machinery of torpedo boats skill, intelligence and experience are essential for securing satisfactory results. Fairly good speeds may be secured very easily, but the purpose of these boats is to run for short intervals at the highest possible speeds, and wreckage of the machinery can be anticipated if a high order of intelligence has not been exercised in every phase of the work, from the designing to the running of the compact, high-powered machinery.

ENGINEERING IN THE NAVY.

By Ira N. Hollis, Member.

An Address delivered before the Naval War College, June, 1903.

The importance of engineering to modern navies is no longer a debatable question. Sails have disappeared and a new propelling power has taken their place so completely that soon there will be few officers of our Service, or of any other Service, who have had any extended experience with ships under sail. Furthermore, within a generation there will be hardly any necessity for such experience. This change from one motive power to another, and, in much of the daily work of the ship from man power to electrical or steam power, is certain to modify radically the point of view of the future naval officer. He will regard that part of the navy connected with the organization and management of ships from a different standpoint.

An historical parallel between the state of the British Navy of the fifteenth and sixteenth centuries and the American Navy during the past fifty years, offers in this connection much that is instructive to naval officers. Both periods saw the disappearance of an organization not adapted to contemporaneous conditions; both closed in a struggle with Spain over colonies in America, and both, for similar reasons, disclosed the woful weakness of Spanish sea power. The series of books on the British Navy published by the Navy Record Society have placed before the general reader much information not hitherto accessible, and among these volumes none is more interesting than that by Sir William Monson.

He wrote a number of tracts on the state of the Navy, and one of them sheds a light not only on the organization for sailing vessels during the Elizabethan period, but it also, by comparison, furnishes food for thought about the future development of modern navies. That part of the tract which relates to

the manning and organization of ships for opposing the Spanish Armada is particularly apt. The history of the decline of Spain's sea power does not seem to have received the consideration it deserved from the point of view of the personnel; and few outside of the Navy understand that when the ships of Spain and England met in 1588 it was a meeting between two different systems, one modern, for that period, and the other antiquated.

Naval warfare, up to the time of *Drake* and *Hawkins*, was conducted on the lines of land warfare, and the ideas of a campaign were similar in both cases. This followed from the fact that the men who commanded the ships had been educated to fight on land. Those who bore armor could not man the oars or the sails of the ship, and their point of view was essentially that of landsmen; hence there was a distinct line of cleavage between the sailors, who navigated and sailed the ships, and the soldiers, who fought. Naturally the latter commanded and bore with them the traditions of their own service. They were not seamen, and they possessed little knowledge of the sea. The development of effective fighting ships was consequently slow, and the true elements of sea power were not thought of, much less understood.

The great voyages of adventure which followed the discovery of the new world bred up a race of seamen fitted to command and to take over from the army the fighting as well as the navigating of ships, and the English were not slow to grasp the advantage to be gained not only from their employment, but from the homogeneity of the crews which would follow when sailors manned the guns as well as the sails. The influence of heavy guns in accelerating this movement was important, but improvements were slow, and two centuries were required to work a complete change in the personnel.

During the latter part of the sixteenth century England passed through a period of reform, especially in the training of officers. Soldiers who could not learn seamanship were eliminated, and their places were taken by men whose previous education and experience had been with the sea and with sails which formed the propelling power. So far as possible all officers and men

were taught to be generally useful on board ship, and there must have been a strong incentive to acquire skill in all the details which related to sails and rigging. Sir William Monson's statement of the case is so clear that a few brief paragraphs are well worth quoting, at the risk of repeating to naval officers much that they know already. He says:

"The experienced valiant Sea-Soldier and Mariner, who knows how to manage a Ship and maintain a Sea-Fight judicially, for Defence of himself and Offence of his Enemy, is only fit to be a Captain or Commander at Sea; for without good Experience a Man otherwise Courageous may soon destroy himself and his Company.

"The Sea Language is not soon learned, much less understood, being only proper to him that has serv'd his Apprenticeship: Besides that, a boisterous Sea and stormy Weather will make a Man, not bred on it so sick, that it bereaves him of Legs, Stomach, and Courage, so much as to fight with his Meat. And in such Weather, when he hears the Seamen cry Starboard or Port, or to bide Aloof, or flat a Sheet, or haul home a Cluling, he thinks he hears a barbarous Speech, which he conceives not the meaning of. Suppose the best and ablest-bred Seaman should buckle on Armour and mount a courageous great Horse, and so undertake the leading of a Troop of Horse, he would no doubt be accounted very indiscreet, and Men would judge he could perform but very weak Service; neither could his Soldiers hope of good Security, being under an ignorant Captain, that knows not scarce how to rein his Horse, much less to take advantage for Execution or Retreat; and yet it is apparent to be far more easy to attain Experience for Land Service than on the Sea.

"The Seaman's desire is to be commanded by those that understand their Labour, Laws and Customs, thereby expecting Reward or Punishment according to their Deserts.

"The Seamen are stubborn or perverse when they receive their Command from the Ignorant in the Discipline of the Sea, who cannot speak to them in their own language.

"That Commander who is bred a Seaman, and of approved Government, by his skill in Choice of his Company, will save

Twenty to the Hundred, and perform better Service than he can possibly do that understands not perfectly how to direct the Officers under him.

"The best Ships of War in the known World have been commanded by Captains bred Seamen; and Merchants put their whole confidence in the Fidelity and Ability of Seamen to carry their Ships and Goods through the hazard of Pirates, Men of War, and the danger of Rocks and Sands, be they of never so much value; which they would never do under the charge of a Gentleman, or an unexperienc'd Soldier for his Valour only.

"The United Provinces, whose Safety and Wealth depend chiefly upon their Sea Affairs, and who for some years past have had great Employment, and enlarged their Dominions much in remote places, use only their expert Seamen to go Captains and Chief Commanders in all their Ships of War and Trade.

"The Spaniards have more officers in their Ships than we: They have a Captain in their Ship, a Captain for their Gunners, and as many Captains as there are Companies of Soldiers; and above all, they have a Commander in the nature of a Colonel above the rest.

"This breeds a great Confusion, and is many times the cause of Mutinies among them; they brawl and fight commonly aboard their Ships as if they were ashore.

"Notwithstanding the Necessity they have of Sailors, there is no Nation less respectful of them than the Spaniards, which is the principal cause of their want of them; and till Spain alters this course, let them never think to be well serv'd at Sea.

"I have heard divers sufficient Men, as Merchants and others that liv'd in Spain, before the Wars with Queen Elizabeth, very much cry down the King of Spain's Ships in comparison of ours; as in particular that they were huge and mighty in Burthen, weak and ill-fashioned in Building, lame and slow in Sailing, fitter for Merchandise than War; and I remember, that old Seamen, as Sir John Hawkins and others, have maintain'd that one of Her Majesty's Ships was able to beat four of them.

"I confess we may the rather believe it, because the Event has shew'd it; for if we examine the particular loss on both sides,

Her Majesty's Ships have devoured divers of the King of Spain's; whereas there was but only one of hers taken, and that merely by the Indiscretion of the Captain Sir Richard Greenville; for which one there have been burnt, sunk, and taken twice, as many as the Queen has in number, insomuch that if the Queen's Loss had equalled the King of Spain's, she could hardly have maintained her Navy in that flourishing state it is in.

"But if we should attribute these Misfortunes to Ships, which are made all of one Fort of Wood and Iron, and after one manner of building, it were great Folly; but give Cæsar his due, and allow the Ships their due, for a Ship is but an engine of Force, us'd for Offence or Defence, and when you speak of the Strength of Ships you must speak of the Sufficiency of Men within her; and therefore, in comparing the Spanish Ships with ours I enter into the Comparison of Men; for if it were in my Choice I rather desire a reasonable Ship of the King of Spain mann'd with Englishmen than a very good Ship of Her Majesty with Spaniards; so much Account I make betwixt the one and the other."

This tells the whole story of the failure of the Armada. Spain was not up to the times. Her men were not familiar with their ships, and they were pitted against seamen trained to handle well the materials placed in their hands.

During the past fifty years history has seemed to repeat itself, and the same change has been taking place in the British Navy. It has only recently culminated in a complete reorganization of the personnel to fit modern conditions, involving precisely the same kind of a readjustment that took place three centuries before. Substitute modern equivalents in engineering for the corresponding technical words in Sir William Monson's paper, and we should have almost a modern essay on the cause of American triumph over Spain.

Our change had begun earlier by the introduction of engineering at the Naval Academy, and in the elimination of subjects unnecessary as belonging to the past, until we had practically an engineering school. Our ships had been gradually organized to promote the highest efficiency of engineering, so that when we met the Spanish ships on which the officers seemed either to

have a contempt for technical engineering matters or no understanding of them, we achieved exactly the same results that the English had accomplished three hundred years before. We found a navy not up to the times, and the result was inevitable from the first.

In the new organization of the English Navy, which has moved very much along the lines begun in the American Navy twenty-three years ago, we find that every officer must enter the service upon the same terms and be prepared after seven years to take a commission in the line, marine corps or the engineer corps. All must learn a large amount of engineering, therefore, before receiving a commission. In the personnel bill which passed Congress we have gone further, and have required that all officers shall be engineers as well as deck officers. The fundamental note in both services must inevitably become engineering, as that of the past has been seamanship. Yet the latter word cannot be lost so long as men go to sea.

It may be useful here to give the ordinarily accepted definition of both words as a means of making clear their exact relation to each other. Engineering is the art of constructing engines or machines, the art of executing civil or military work which requires a special knowledge or use of machinery or the principles of mechanics. Seamanship is an acquaintance with the art of managing and navigating a ship at sea.

It is difficult to decide which of these two words includes the other when applied to the naval service. In a battleship both involve the handling and training of bodies of men. Perhaps, since a battleship can be managed only by means of machinery, engineering is legitimately a branch of seamanship. It is a branch of overwhelming importance; but, nevertheless, there are many other things besides the management of a ship involved in seamanship. A knowledge of the sea and of storms might very properly be classified as seamanship. Still, engineering, as involving the motive power, the guns and the technical qualities of the ship, is the chief thing to be studied.

During the discussion of the personnel bill one of the advocates of the bill referred to the future naval officer as a fighting

engineer, and in many senses the description is a correct one, for engineering must necessarily be the motive, or the foundation, upon which the education of a future naval officer is built up. The change recognizing this fact has already taken place at the Naval Academy. The personnel bill, therefore, came as a final settlement of a struggle which had marked the transition from sails to machinery. It formed a clear dividing line between a ship which could keep the sea for an indefinite period and one which had only a steaming radius dependent upon the coal supply. The old type had lasted for about four centuries, having succeeded still another kind of vessel which had a radius dependent upon the muscular power of rowers. In some respects coal endurance is similar to muscular endurance; both may become exhausted, and both are limited to definite distances from the home base.

Whatever naval officers or civilians may think of the merits of the personnel bill, nothing can ever carry us back to the old regime which prevailed in the unsatisfactory union of sails and steam within the same vessel. The change from one type to another was therefore necessarily accompanied by a change in the education of officers and men, in the organization, and even in the strategy and tactics adopted in the time of war. One of the most important problems which our Service has to solve today is the relation of the present engineering to all three of these points.

It is difficult to write on engineering in the Navy without repeating much that has already been said. Every officer knows that the modern battleship is directed in almost every detail by machinery, and that the organization must be such as to secure the highest efficiency of the machines. Some are used for moving the ship as a whole, some for handling the guns and turrets, and some for the numerous auxiliary purposes connected with the comfort and convenience of the crew. Few can be spared, and in many cases a single weak element would jeopard the safety of the entire ship's company. To the end that every machine, including the guns which, after all, are but special types of machinery, shall perform the function allotted to it in its original design, every man must be well trained in the care and use of

machines or boilers. The responsibility of this training falls upon the commanding officer, who must therefore have an intelligent understanding of every element which affects the efficiency of his ship; otherwise he cannot do his whole duty without undue effort and anxiety.

It is not sufficient that a man should have common sense and experience of the sea in order to enable him to take good care of a battleship. He may be able to carry her out of port and to bring her back again. He may have a good knowledge of international law, and therefore be able to serve the nation well in diplomacy. He may, furthermore, be able to discipline his crew; but in the test of a long war, where preparation and care of machinery are bound to count in success or failure, he is badly handicapped if he does not know from personal experience all the details of his ship. His early education must, therefore, have been related to the material problems of his command, and his experience must have been such as to fix definitely what he has learned at school.

The Naval Academy is no longer a school of seamanship, unless we define seamanship in a very broad sense, but it is essentially a school of engineering, where all the fundamental subjects are such as would be found in a good technical school. The course of study for all cadets today is along much the same lines as that given to cadet engineers twenty-nine years ago, only it is a better engineering course. Graduates can now fill engineers' positions in civil life quite as well as young men from technical and scientific schools. This seems not to be well understood even by the Service, and its effect in modifying the entire personnel of the Navy, as class after class graduates, cannot be estimated. Of one thing we may be sure, namely, that in the management of ships and in the great number of details connected with their design, their equipment and their organization, the engineering motive is bound to replace the traditions which formed themselves around the sails and the broadside guns of the old frigates. This has up to this time been recognized only in part by the Navy, where the benefit of the personnel bill seems still open to argument.

My own correspondence and discussion with officers would lead me to say that there are two parties in the service; the larger has accepted the situation, and the smaller is disposed to question seriously the whole change. In the eyes of the former the management of the Navy under the new organization has been entirely successful, while the latter believe that a new engineer corps will be established within a comparatively short time. It seems to me that the personnel bill has not yet had a fair trial. While much has been done at the Naval Academy, the ordinary routine on board ship has not adapted itself to the continuation of the work done there, and the obvious intention of the bill has not been thoroughly carried out. One familiar with the Navy knows perfectly well that there have been difficulties in the way. Every bureau has been clamoring for officers, and every corps has felt itself depleted during the past five years. We have had this great reorganization to carry out at a time of tremendous expansion in colonial interests and in ships. There have not been sufficient officers to go around, and therefore no branch of the Service has had its work done to the best advantage. Nevertheless some method should have been found in the Service to provide successors for the old engineer corps and for those skilled in engineering who are rapidly disappearing from the active list. Within ten years there will be few officers available for sea service who in the early years of their careers had responsible engineering duties.

It is not enough to detail a young graduate of the Naval Academy to receive instruction in the engine room; he must have responsible charge, and there is no reason why he should not be capable of taking it a few months after graduation. This should not be a question of the wishes of any individual, either subordinate or in command, but it should be the absolute requirement for a young officer's promotion. The theoretical objection that officers cannot alternate between the engine room and the deck all their lives, does not apply at all, as the personnel bill never contemplated anything of this kind. It did contemplate, however, that the care and running of machinery should be placed in the hands of officers of the line, and that, therefore,

they must obtain experience early in their careers. Three years in the engine room is none too long, and these three years should be within the first six years out of the Naval Academy. After that, officers should be detailed according to their tastes and abilities.

In discussing the engineering question officers commonly confuse the design and construction of machinery with its care and management in service. The two have no fixed, definite relation. The best designer of a gun, or of an engine, may not be a good man to manage it in actual service. For the work connected with design and, to a large extent, with repairs, a technical department of the line, or a separate technical corps, will be absolutely necessary, and there is nothing inconsistent with such a corps in the personnel bill. The intention of that bill was to end the strife in the Navy and to force engineering into the daily duties of line officers. The hitch in the old organization was the total absence of inducement to study engineering, except in the Engineer Corps. So long as a new Engineer Corps is chosen from the line after officers have had sufficient experience to know their own tastes, the spirit of the new organization will be maintained. Furthermore, it is doubtful if there is anything in the statutes to prevent the Secretary of the Navy from detailing officers permanently to engineering duties, and thus, by administrative process, creating an Engineer Corps. The main difficulty in the way would be found in securing the willing coöperation of officers so detailed unless some inducement were held out.

When we consider the care and running of machinery, too much has been made of the phrases that "this is an age of specialization," and that the present scheme is "contrary to the genius of modern times." That is perfectly true so far as it goes, but on board ship a good officer is the specialist, and the more he knows about his ship the better for the Service. As a matter of fact, there is no difficulty whatever in teaching a midshipman all the details on board ship if the opportunities are placed before him. With his grounding in engineering he is ready to take hold of anything and to learn the management of men besides.

The fundamental fact to be kept before him at all times is that the care and adjustment of the machinery of a ship, from the propellers to the turrets, demand greater judgment and energy than running it. If a machine is in good order, pulling a lever ought to start it or stop it. Young officers should, therefore, have a large share in duties connected with repairing, cleaning and adjusting machinery and boilers, and too much emphasis cannot be placed on the fact that this share must be a responsible one.

There is danger that the Service may drift into too great a dependence upon the warrant machinists, bringing up again within a few years another discussion similar to that which prevailed for thirty years after the Civil War. This would be a most unfortunate condition of affairs, and all the work connected with organization would have to be done over again. What could have been thought of a system which turned the practical management of sails over to the warrant officers under the theory that the sails and rigging formed only one of the many details under the care of executive officers? The commanding officer would soon have found himself as much in the hands of warrant officers as were the soldiers who commanded before the reorganization of the British sailing navy, and as will be the officers of our Service in case the machinery is allowed to fall back into the hands of officers without commissions. The opening of the warrant grades to machinists was intended to promote efficiency by giving them the same incentive already provided for the other enlisted men. It would be most unfortunate to have it used as a means of relieving line officers of watch duties below decks. No doubt, there was good reason twenty years ago to question the wisdom of putting five or six highly educated engineers on board of a sailing vessel with auxiliary steam power to run engines of a few hundred horsepower, but that is not the state of the Navy today, and we must not forget that everything connected with the materials of a ship depends more or less directly upon the efficiency of machinery. The most important consideration cannot be allowed to drift

back into one of secondary importance, as it could have been when vessels could keep the sea by means of their sail power.

When we consider the possibilities of a technical corps, we should find it difficult to make an argument against it, and it could not in any way decrease the effectiveness of the present reorganization, engineering being considered the motive underlying the education at the Naval Academy. In fact, a technical corps, or a technical department of the line, is almost a necessity, but the men for this department should be selected from the list of line officers, and they must have developed technical abilities. Because all officers have been educated alike, it does not follow that they are equal in the performance of duties, or that all will have the same tastes, and it would be a serious loss if the Government could not take advantage of special talent of its commissioned officers. If this looks like specialization, it is the kind that selects men for the work that they are best fitted to do after they have demonstrated their fitness. Such specialization will not throw men out of touch with the executive duties of a ship, as the care and management of machinery demands the co-operation of a large number of enlisted men, who must be directed.

This kind of control on the part of the chief engineer and his subordinates involves the same class of executive work to be found on deck, and an officer whose principal occupation has been with machinery is not thereby crippled for executive duties. The old system of detailing men to three or four years' clerical work in the Navy Department or in the yards was far worse on the morale of the Service than would be a long detail to engineering duty at sea or on shore. Far better that men should have been attending to construction and engineering matters on shore than that they should have been hanging over desks.

In the organization and training of the crew the problem is more difficult. The Navy cannot keep in the service first-rate mechanics, as the rewards in civil life are too great. With few exceptions, therefore, we have to be content with men of the ordinary laboring class. By making their duties simple, and offering suitable rewards, they can be trained to do the work of a battleship admirably. These are matters, however, which

experience must arrange for the naval service, and there is no reason to think that naval officers are not amply qualified to adjust them properly. Still, one of the phrases which has been worked to death in the interest of the men who customarily man the guns is "The man behind the gun." This phrase tends to set up a caste distinction in the Service which has no existence in fact. It puts a premium on the spectacular, and tends to destroy the spirit of those men who do the work of preparation and who carry the gun into a position where it can be fired. Everyone knows that a battleship is built for her guns, and that she is nothing if not a first-rate gun carriage, but that does not entitle the men who stand in the turrets to the peculiar gratitude of the Nation when compared with the men who tend water, hoist powder, or who handle the throttle valves of the engines. As a matter of fact, all the crew are behind the guns.

As a nation we are likely to go into hysterics over the spectacular and for the time being to lose sight of essentials. The great voyage of the *Oregon* from San Francisco to the Atlantic coast will probably live forever in the annals of our Navy, and yet many other cases of fine service have entirely escaped the attention of people. Whoever speaks or cares anything about the exploit of the little *Marietta* that accompanied the *Oregon?* No one seems to have thought of her plight in case she had met the Spanish fleet in company with the *Oregon.* Her business was to go ahead and order coal and to act as a scout for the *Oregon.* She made the same voyage and with equal success, yet we have been so drilled to admire the spectacular that we forget her entirely. Furthermore, who thinks much about the voyage of the *Monterey* across the Pacific?—as fine a piece of seamanship as we have had in modern times. It is too often the actor in the middle of the stage who catches our attention, no matter whose lines he may be reciting.

All this is an important consideration in organizing the crew. Every man should have full recognition for good work, and it is especially important that technical skill should have its reward. I am convinced that the education of deck officers into a sympathetic understanding of the machinists, oilers and firemen will

more than offset any deficiency that may arise from their lack of professional contact with the machinery.

In writing about organization we necessarily draw from a very limited experience with steam vessels in naval warfare. There has been no real test. The Civil War gave us great experience in the conduct of a blockade, but none whatever in great fleet tactics. Our late war, although large in results, taught few lessons outside of technical considerations connected with the design of ships, and we have little upon which to pass a judgment of the future. Naval tactics and strategy may have as great a development under steam as they had under sails, given the presence of a great seaman engineer. Some writer has defined sea endurance as the essence of naval strategy, and steam power as the essence of naval tactics. This seems a clear and apt distinction which may well be kept in mind. They depend respectively upon the economical use of coal and the proper use of materials for handling a ship. Neither has received the attention its importance demands. Much can be done in studying the economical use of coal for our battleships and cruisers at high and low speeds, and the Department could probably extend the steaming radius of every ship by the careful elimination of unessentials. and a more careful study of every type of ship.

The only great lesson taught by the Spanish War is the enormous advantage of adequate preparation. We already knew that. No officer in the Navy had the slightest doubt about the ultimate outcome of the war, even from the day of its declaration, and for probably the same reason that would have governed his judgment of the great Armada. The Spaniards were as much behind the times of 1898 as they were in 1588, and their antagonists were as much ahead. Nevertheless, we now bid fair to drift, having accomplished a great reform under conditions which have led the greatest navy in the world to copy us. We do not yet seem to realize our opportunities and duties.

Naval warfare of the future cannot possibly be a matter of long campaigns. A great fleet action will be short and sharp, and few or none of the defeated ships are likely to escape. Maneuvering for the weather gauge will be replaced by the long and tedious

preparation of bringing every ship up to the supreme test of one great effort, and by the attempt to get between an enemy's fleet and his coal pile.

The future of our Navy is fortunately in the hands of men with a natural aptitude for mechanics and engineering, and in this period of transition we have the advantage of no real lack of harmony to prevent an efficient organization. The country owes much to the patience and courage of the older officers who were educated under a system which has already disappeared. Upon their shoulders have been thrust some very heavy burdens. Men advanced in life cannot easily learn new ways, and yet our officers have risen to the demands of a new service with admirable courage. Instead of repining over the unsatisfactory settlement of a great question, the entire Navy ought to rejoice that so few mistakes have been made, thus giving positive evidence that the Service will be equal to the occasion. The main point under consideration here is that we should not wait too long to provide successors for a fast disappearing corps.

A COMPARISON OF THE PERFORMANCES OF THE MONITORS *ARKANSAS, FLORIDA, NEVADA* AND *WYOMING.*

By Assistant Engineer R. E. Carney, U. S. N. (Retired), Member.

An extended comparison of the performances of the four single-turretted monitors built under the Act of Congress, May 4, 1898, would be interesting were there more absolute data on which to make such comparison. As known to the Journal's readers, from accounts of their official trial performances as previously published, these vessels are identical in hulls and practically so in engines, the chief points of difference being in the character of boilers installed and the design of propellers fitted to the shafts.

As regards their boiler installations, their general characteristics are as follows:

Ship.	Type of Boiler.	No.	Total H. S.	Total G. S.	Ratio H. S. to G. S.
			sq. ft.	*sq. ft.*	
Arkansas..	Thornycroft.	4	9,368	198	47.31 to 1
Florida....	Mosher.	4	9,600	240	40 to 1
Nevada....	Niclausse.	4	8,874	220	40.336 to 1
Wyoming.	Babcock & Wilcox	4	8,800	200	44 to 1

The boilers of the four vessels are designed for 250 pounds pressure, with a reducing valve between boilers and engines, so as to secure at the engines a pressure of 200 pounds. The *Arkansas* and the *Nevada* have athwartship, while the *Florida* and *Wyoming* have fore-and-aft, fire rooms, all the boilers in each vessel being in one watertight compartment.

The engines being practically identical in design, their performances are not of particular interest. The amount of steam used by the engines of the four vessels may be stated, as roughly

deduced from average cards taken during the full-power trials, to have been respectively as follows:

Arkansas, pounds per hour, . . .	34,600
Florida, pounds per hour, . . .	34,000
Wyoming, pounds per hour, . .	32,000
Nevada, pounds per hour, . . .	25,000

These amounts are not reduced to equivalent values from and at a temperature of 212 degrees from lack of necessary data to make such reduction. They are worked out from the points of release on the L.P. cards and take into consideration the clearances. Omitting this, however, in the comparison, the Mosher boiler has the advantage of all the others in lightness.

The total weight of machinery (including boilers) of the *Florida* is 28.28 tons less than that of the *Wyoming*, 30.22 tons less than that of the *Arkansas*, and 55.35 tons less than that of the *Nevada*. The difference of weights of the boilers alone of the *Arkansas* and *Florida* is 56,081 pounds, which coincides closely with the total difference of weights of installations, and it may be presumed that the other differences are likewise due to boilers alone.

Taking the actual boiler performances on trial, we find that per actual H.P. developed the Thornycroft boilers required 5.184 square feet, the Niclausse 4.428 square feet, the Mosher 4.008 square feet, and the Babcock & Wilcox 3.589 square feet of heating surface, placing the latter boiler first in efficiency of heating surface, the Mosher second, the Niclausse third and the Thornycroft fourth.

The consumption of coal per H.P. developed was: For the Thornycroft 3.719 pounds per hour, for the Niclausse 2.6505 pounds, for the Babcock & Wilcox 2.421 pounds, and for the Mosher 1.986 pounds per hour. The advantage of the Mosher boilers in this particular may be questioned, from the fact that the officer in charge of the fire rooms during the trial noted that the fires were burned down during the latter part of the trial, instead of being kept to the level at which they were at the beginning of the trial. If during that time the fires had become one

inch less in depth over the 240 square feet of grate, it would require half a ton of coal to bring them to a proper level, and this amount additional would bring the consumption up to 2.22 square feet per hour per H.P. developed. Unfortunately there is no definite knowledge as to the length of time during which no firing was done, but it seems not probable that the consumption was less than this latter rate. The coal burned per hour per square foot of grate would indicate that the coal consumption of the Mosher boiler was incorrect as reported, more particularly when it is remembered that the average air pressure used was .5 inch for the Thornycroft, .417 inch for the Mosher, and .375 inch for the Babcock & Wilcox boilers, the *Nevada* alone working under natural draft, which was assisted by the blowers during the last fifteen minutes, as much as the open fire rooms admitted.

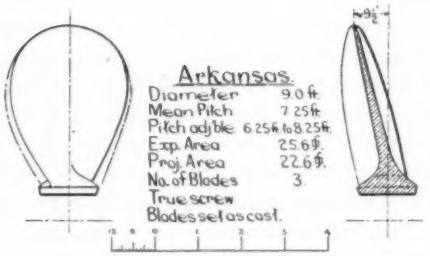
The amounts burned per hour per square foot of grate are:

Thornycroft, square feet, ...	33.94
Babcock & Wilcox, square feet, . .	29.68
Niclausse, square feet, ...	24.14
Mosher, square feet, ...	19.82

Even at an assumed consumption of 2.22 pounds per hour per horsepower developed, the Mosher boilers' consumption would be only 22.65 pounds per square foot of grate per hour, which seems low for the air pressure under which the trial was made.

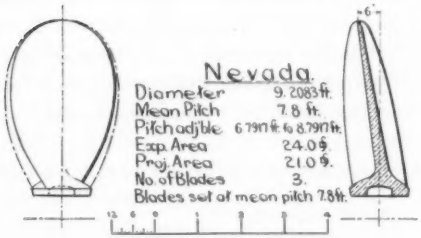
In summarizing the above, the only apparent basis on which steam production can be compared would seem to be the total horsepower developed, in which case the Babcock & Wilcox boiler would be first, the Mosher second, the Niclausse third and the Thornycroft fourth; but this basis of comparison must not be given too much value, as both the *Arkansas* and *Wyoming*, on their standardization trials, considerably exceeded the power developed on their speed trials. The true comparative values of these four types of boilers will only be brought out by the results of their actual performances during continuous years of service.

Remembering that the work of the engines consists in main-



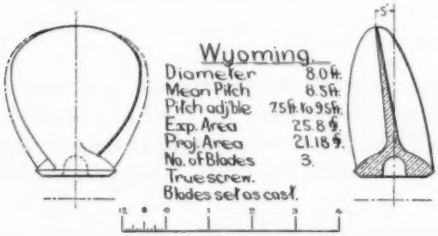
Arkansas.

Diameter	9.0 ft.
Mean Pitch	7.25 ft.
Pitch adjble	6.25 ft. to 8.25 ft.
Exp. Area	25.6 ☐ ft.
Proj. Area	22.6 ☐ ft.
No. of Blades	3.
True screw.	
Blades set as cast.	



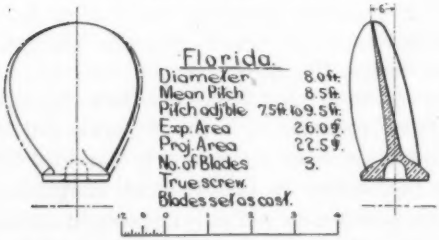
Nevada.

Diameter	9.2083 ft.
Mean Pitch	7.8 ft.
Pitch adjble	6.7917 ft. to 8.7917 ft.
Exp. Area	24.0 ☐ ft.
Proj. Area	21.0 ☐ ft.
No. of Blades	3.
Blades set at mean pitch 7.8 ft.	



Wyoming.

Diameter	8.0 ft.
Mean Pitch	8.5 ft.
Pitch adjble	7.5 ft. to 9.5 ft.
Exp. Area	25.8 ☐ ft.
Proj. Area	21.18 ☐ ft.
No. of Blades	3.
True screw.	
Blades set as cast.	



Florida.

Diameter	8.0 ft.
Mean Pitch	8.5 ft.
Pitch adjble	7.5 ft. to 9.5 ft.
Exp. Area	26.0 ☐ ft.
Proj. Area	22.5 ☐ ft.
No. of Blades	3.
True screw.	
Blades set as cast.	

PROPELLERS HARBOR DEFENSE MONITORS.

taining in the shafts sufficient torque to overcome the angular resistance to the rotation of the screws, and recognizing the fact that the engines are so nearly alike that they would probably develop practically similar results under like conditions, it is not considered necessary to specifically analyze the engine performances.

It is in the performance of the screws of these four ships that the greatest interest centers, for, as is well known, the Bureau of Steam Engineering allows the contractors considerable latitude in this respect, since they are permitted to fit their own design of screws, as the builders alone are responsible for the performance of a ship on trial. In the present case there was a wide divergence of practice upon the part of the four firms. A summary of the screw data is given below, showing how varied can be the conditions assumed to reach a required result:

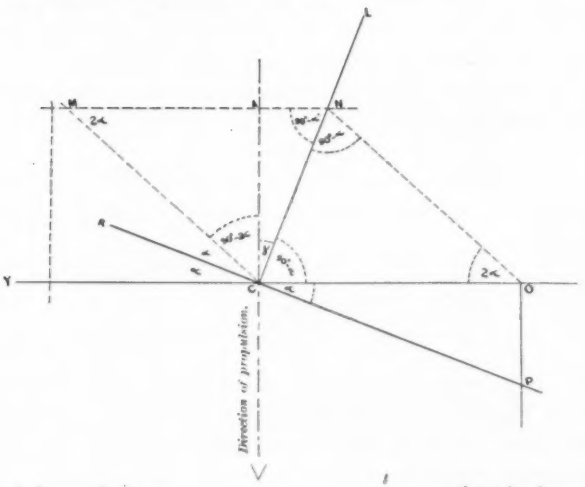
Name of ship	*Arkansas.*	*Florida.*	*Nevada.*	*Wyoming.*
Screw designed by	Newport News S. B. & D. D. Co.	Crescent Ship Yard.	Bath Iron works.	Union Iron Works.
Number of blades	3	3	3	3
Diameter, feet and inches	9-0	8-6	9-2.7	8-0
Pitch variable from, feet and inches,	6-3 to 8-3	6-9 to 8-9	6-6.83 to 7-9.55	7-8.6 to 9-4.4
as set on trial, feet and inches,	7-3	8-3⅛	7-9.55	8-5.9
Helicoidal area, square feet	25 6	26	24	25.8
Projected area, square feet	22.6	22 5	21	21.18
Pitch ratio, $\frac{P}{D}$	.8055	.9797	.8452	1.0613

The mean pitch as given on the sketches (page 826) is the mean of the two limits of adjustment, and not the mean pitch of the screws as set on trial, which latter is used in the calculations in this article.

The solution of propeller performances is more or less indefinite, being largely theory, and depending on assumptions which may or may not be correct.

In trying to compare the performances of the above four propellers, I have made use of a method which is no doubt old, and have assumed in doing it that the mean pitch is applicable, and that the forces exerted are proportional to the resulting movements. Even the proportion of the engine power actually transmitted to the screw has not, as far as the writer knows, been accurately determined in any case, though experiments to determine it were interrupted at the Naval Academy by the breaking

out of the war with Spain. I have further assumed that the total driving force exerted by each blade may be considered as concentrated at the tip of a radius which is the mean of all the radii of infinitesimal portions of the blade, or, rather, I have taken in this case a radius the arc of which would divide the blade into two equal parts, one outside and the other inside of the arc. The determinations of these radii were made from reduced drawings, and may not be entirely accurate, though as close as could be determined from a small scale. Calling this radius the "effective radius," it is found to be for the



Arkansas' screw, ...	36.00 inches.
Florida's screw, ...	30.15 inches.
Nevada's screw, ...	30.84 inches.
Wyoming's screw, ...	30.21 inches.

Referring to the diagram, the line CO, in the plane of rotation, represents the developed circumference of the effective radius which is equal to its movement in the plane of rotation. OP represents the mean pitch, and both these dimensions are in feet. Then the driving face of the propeller will be contained in the plane perpendicular through RCP, and the angle a, which may be

called the "pitch angle," $= PCO$, is known from its tangent which equals $\frac{PO}{CO}$. Since the resultant of all currents driven sternward from the propeller will be in a line perpendicular to its face, these may be represented as extending along CL. Considering only one revolution of the propeller, and regarding the latter as the reverse of a turbine, we know that whether the movement is from L to C, or from C to L, corresponding operations ensue. Knowing that the blade in its revolution runs through CO at an angle α to its face, and that the final resultant of all the driven currents is along CL, it follows that a movement along CM, also at an angle α to the face of the blade is required to balance CO, in order that the resultant shall travel along CL. This gives us two sides of what may be called a parallelogram of forces—though so far it is of movements—and by drawing parallels to each side through the terminal of the other we find that these parallels intersect each other and CL at N and the parallelogram is completed. The values of the different angles are easily determined from that of the one known angle α, but need not be determined, as all the desired values can be trigonometrically worked out in terms of two of the known quantities.

Of the resultant currents driven sternward, CN, only the fore-and-aft component CA is useful in propulsive effect, the transverse component AN being wasted in giving athwartship motion to the water. Incidentally, it is this component, or rather its reaction, which sets up a torsional strain in the shafting in an opposite direction to that imposed by the varying turning moments of the engine.

Now in the triangle CNO, in which we have two known quantities, the angle α and the side CO, the other two angles may be easily determined.

Having learned their values in terms of α, we have in the above named triangle

$$\frac{\sin (90° - \alpha)}{\sin 2\alpha} = \frac{CO}{CN},$$

whence

$$CN = \frac{CO \sin 2\alpha}{\sin (90° - \alpha)} = \frac{CO\ 2 \sin \alpha \cos \alpha}{\cos \alpha} = 2\ CO \sin \alpha,$$

and since in the triangle ACN, $AC = CN \cos a$, we have by substitution

$$AC = \begin{cases} CO \sin 2a, \\ 2CO \sin a \cos a, \end{cases}$$

and similarly

$$AN = 2CO \sin^2 a.$$

Using values of CO equal to the developed circumferences of the "effective radii" of the four screws, which give values of 18.55′, 15.79′, 16.15′ and 15.82′ respectively for the screws of *Arkansas*, *Florida*, *Nevada* and *Wyoming*, and taking values of $OP = 7.25'$, $8.32'$, $7.8'$ and $8.49'$, the pitches of these screws on trial, from which the angle is found to be for the *Arkansas*' screw 21° 20′ 52″, for the *Florida's* 27° 27′ 7″ for the *Nevada's* 25° 46′ 45″ and for the *Wyoming's* 28° 13′ 14″.

Placing the values deduced in tabular form, we have for one revolution—

.....	*Arkansas.*	*Florida.*	*Nevada.*	*Wyoming.*
CO.....	18.55′	15.79′	16.15′	15.82′
OP.....	7.25′	8.32′	7.8′	8.49′
a.....	21° 20′ 52″	27° 27′ 7″	25° 46′ 45″	28° 13′ 14″
CN.....	13.508′	14.560′	14.047′	14.961′
CA.....	12.578′	12.9208′	12.694′	13.189′
AN.....	4.9162″	6.7129′	6.1094′	7.2393′

Having found later that the performances of the *Arkansas* and *Wyoming* on their sea trials gave results not compatible with those over the measured mile, and on the assumption that the error was due to a slightly incorrect estimate of the speed as derived from the speed and power curves of these ships, I have taken for these two vessels their highest speeds over the standardization course in the following determination of the distribution of power by the screws, and have interpolated their performances on the sea trials in their proper places, with the results accomplished on the standardization runs. Evidently the fore-and-aft component CA will be composed of two elements—the advance of the ship or screw per revolution, and the reverse current. The first is obtained, of course, by dividing the speed of the ship per minute by the revolutions per minute,

and the reverse current will be the remainder left after subtracting the above from CA. This gives

.....	*Arkansas.*	*Florida.*	*Nevada.*	*Wyoming.*
CA =.....	12.578	12.9208	12.649	13.189
Advance of screw.....	6.0058	6.3115	7.0538	6.0141
Reverse current.....	6.5722	6.6093	5.5952	7.1759

Since the two elements of CA given above represent the projections on the fore-and-aft line of the portions of CN devoted to producing them, as AN represents the projection on the athwartship line of the portion wasted, by projecting these quantities back on CN perpendicular to the latter, or determining the values of such projections, we can get the proportions of the resultant current along CN devoted to each of the several motions, and on the assumption that these motions are proportional to the forces producing them, we can get an idea of the distribution of power by the screw, remembering, however, that this represents an unknown fraction of the power of the engines. The proportionate values of the projections of the different motions on CN to the total value of CN can be ascertained by substitution in the terms

$$\frac{\text{advance of screw} \times \cos \alpha}{CN}, \frac{\text{reverse current} \times \cos \alpha}{CN} \text{ and } \frac{AN \sin \alpha}{CN},$$

and by making these substitutions, and assuming that they represent the forces instead of the motions, we find the percentages to be as follows, taking the nearest whole percentages, viz:

.....	*Arkansas.*	*Florida.*	*Nevada.*	*Wyoming.*
	per cent.	*per cent.*	*per cent.*	*per cent.*
To advance of screw (of ship)...	42	39	45	35
reverse current.....	45	40	36	42
wastage athwartship.....	13	21	19	23

According to Rankine, that propeller is best which drives the largest volume of water astern at the least speed, and to make a comparison in the above cases it would seem proper to reduce the movement of the reverse current to a time basis of one second's duration and multiply the reverse current so derived

by the helicoidal area of each screw for a comparison. This would give us as results, in cubic feet of water per second for one screw, the following for the several ships:

Nevada, ...	419.804
Arkansas, ...	504.164
Florida, ...	569.862
Wyoming, ...	663 714

This would indicate the superior efficiency of the *Nevada's* screw, that of the *Arkansas* as second, *Florida* third and the *Wyoming's* as the poorest.

The performance of the screws of the *Arkansas* and *Wyoming* on their standardization runs, with the performances of the trial runs interpolated, are given below — at least the information making up the fore-and-aft components of the screw thrust.

Speed.	Revolutions.	Advance of screw.	Reversed current.
knots.	*ARKANSAS.*		
4.308	75.25	5.9344	6.6436
8.4	120.68	7.0536	5.5244
10.591	165.44	6.4873	6.0907
(*a*) 12.4	197.1	6.536	6.042
12.7125	214.5	6.0058	6.5722
	WYOMING.		
5.97	91.05	6.612	6.557
8.57	134.02	6.4715	6.7175
10.04	170.5	6.1813	7.0077
(*a*) 11.08	201.3	5.9403	7.2487
12.37	214.7	6.0141	7.1741

(*a*) Trial speeds.

From Rankine's view, taking the proportions of the fore-and-aft component AC devoted to the reverse current, it will be seen that the *Arkansas'* screws were most efficient at a speed of about 8.4 knots, and that there was a decrease in efficiency both above and below that speed. The screws of the *Wyoming* were most efficient at the lowest speed. The apparent discrepancies in the values shown for the trial speeds, which are apparently above the real efficiency in the case of the *Arkansas* and below it in the case of the *Wyoming*, are attributed to the fact that the stated speeds were probably taken from the speed curve. The stand-

ardization points were secured from the ships running over a measured course, and therefore can be considered much more reliable.

As a propeller with its driving faces parallel to the shaft would simply rotate the water without producing any forward motion, and a propeller with faces perpendicular to the shaft would rotate without any driving effect, there must be some angle of the blade between these positions—that is between 0 and 90 degrees—which will produce the greatest desired propulsive effect in proportion to the power exerted.

When the angle α equals 45 degrees, equal amounts of the energy expended will be devoted to propulsive and wasted transverse effects, while as α expands beyond 45 degrees the proportion of power wasted increases up to $\alpha = 90$ degrees, when all the power is wasted. Similarly, as α decreases from 45 degrees toward 0 degrees, the sides of the parallelogram ON and CM approach nearer and nearer to the horizontal, and both AC and the proportion between AC and AN become less, while they finally reach zero. The maximum effect for a given power, taking into consideration both useful and wasted efforts of the screw, is reached when $\alpha = 22$ degrees 30 minutes, at which time 85 per cent. of CN is devoted to driving effect divided between advance of screw and reversed current, and 15 per cent. is wasted in setting up transverse motion in the screw current. Of course, the element "advance of screw" or "advance of ship" will depend on the form of the hull, which governs both hull resistance and increased speed of wake, and the element of "reversed current" will depend on the former. The latter will, however, be influenced in amount by both its speed and the helicoidal area of the screw, and the successful propeller will be one which, with a large area, will have the greater ratio of "advance of screw" to "speed of reverse current."

For a pitch angle of 22 degrees 30 minutes, in a propeller of diameter D, and effective radius $= x\frac{D}{2}$, the proper pitch is expressed by $p = 1.3012\,Dx$, and the pitch ratio by $\frac{P}{D} = 1.3012\,x$.

THE DEVELOPMENT OF THE YARROW BOILER FOR USE IN BATTLESHIPS.

By William Crush, Manager, Boiler Department, Yarrow & Co., Ltd.

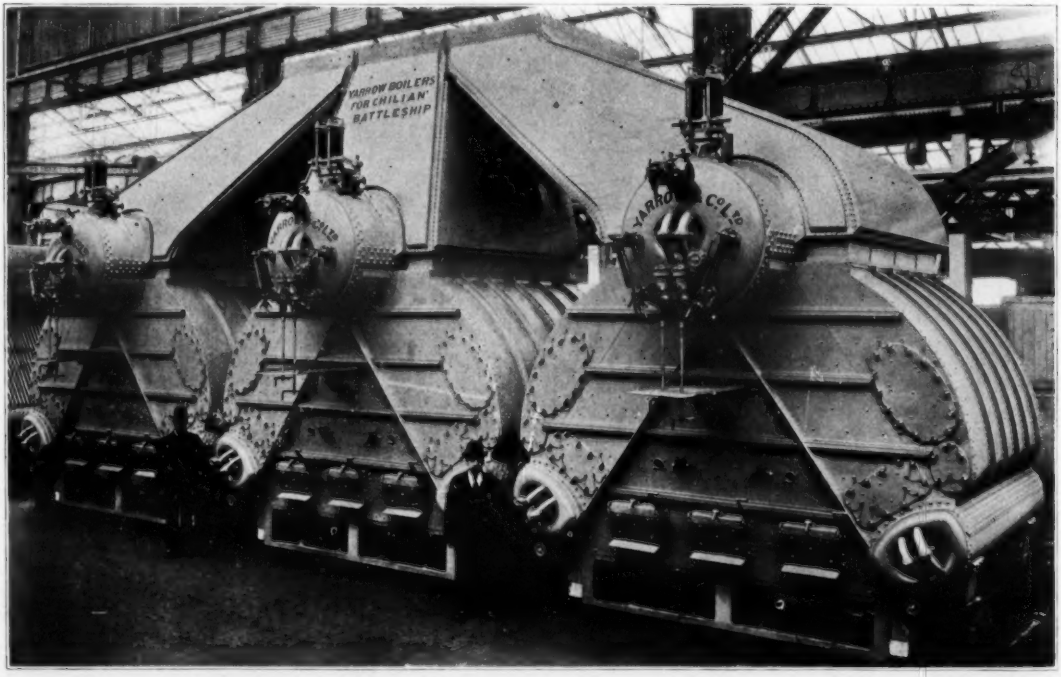
In January, 1903, there were, either under construction or in commission, thirty-four battleships and cruisers intended for service in the British, Chilian, Austrian, Dutch, Swedish, Norwegian and Portuguese navies, all fitted with the Yarrow boilers, and representing upwards of 308,700 indicated horsepower. This is exclusive of the installation of this type of boiler in destroyers, torpedo boats and gun boats.

Referring to the above, it is interesting to trace the growth and development of the Yarrow boiler from 1887 to the present day.

ITS INTRODUCTION INTO THE BRITISH NAVY.

From 1887 to 1892 Messrs. Yarrow & Co. had been so uniformly successful in their efforts to adapt this boiler for torpedo boats, high-speed steam yachts, and similar craft, that the British Admiralty commissioned them to design and build two fast torpedo-boat destroyers, which were the first of that class of vessel ever built, and were named respectively *Havock* and *Hornet*.

With a view to test the Yarrow boiler on a larger scale than had hitherto been attempted, the *Hornet* was designed and fitted with this type of water-tube boiler, while in the *Havock* was installed the locomotive type of boiler. The two vessels, except for boiler installations, were in every respect identical, and therefore provided valuable means for securing comparative data as to the relative value of the two types of boilers for naval purposes. It may, therefore, be said that with H. M. S. *Hornet* the adoption of the Yarrow boiler for sea-going vessels commenced.



YARROW WATER-TUBE BOILERS FOR TWO CHILIAN BATTLESHIPS. TWELVE OF THESE BOILERS IN EACH VESSEL. I.H.P., 12,500.

So successful was the *Hornet* as a torpedo boat that the British Admiralty, recognizing the new type of vessel as one likely to be invaluable in naval warfare, immediately placed with shipbuilders throughout Great Britain orders for many similar vessels, ten of which were fitted with Yarrow boilers.

The attention of naval authorities throughout the world was thus specifically directed to the boiler question for torpedo boats, and as a result the Imperial Russian Government commissioned Messrs. Yarrow & Co. to design and build for them a vessel of an improved *Hornet* type. This vessel was called the *Sokol*, and was at that time the fastest sea-going vessel in the world, since she attained with all trial weights on board over 30 knots speed.

In passing, it is worthy to note that the *Sokol* marked yet another important development in naval progress, that of the introduction of high tensile steel in the construction of torpedo-boat destroyers, which material has since been adopted for this class of vessel.

ITS ADOPTION BY CONTINENTAL NAVIES.

About this time the Dutch naval authorities had under consideration the designs of three first-class cruisers, the *Holland*, *Zeeland* and *Friesland*, vessels of 3,900 tons displacement each, and of 10,000 H.P.

There are few naval powers more progressive in policy than the Dutch. Efficiency of the highest order prevails throughout the service, and those who direct its naval policy combine the advantages of a wide practical experience and high scientific attainments. The naval authorities determined upon fitting these new cruisers with a combination of water-tube and marine boilers, consisting of eight Yarrow boilers and two cylindrical boilers, 7,500 H.P. in Yarrow boilers and 2,500 H.P. in cylindrical boilers.

So successful were these vessels that, seven years later, the British Admiralty followed the example of the Dutch naval authorities and adopted the principle of combined boiler installation in the "County" class of cruisers. The *Hampshire* and *Antrim*, now in course of construction, are thus fitted with Yarrow boilers in combination with cylindrical boilers.

The Yarrow boiler is now adopted in the Dutch navy in six cruisers and three battleships, providing a total of 76,500 horse-power of boilers of this type in that service.

Contemporaneously with the installation of the Yarrow boiler in the Dutch navy, the Austrian naval authorities adopted it in their cruisers *Zenta*, *Aspern* and *Szigetvar*, and later still in one more cruiser and three battleships.

Among the other navies where the Yarrow boiler has been adopted with most satisfactory results is the Swedish service. This boiler has been installed in the following vessels of the navy of that nation, the total H.P. of the vessels being 52,500:

Aran,	*Psilander*,	*Wasa*,
Class *Uggla*,	*Dristigheten*,	*Tapparheten*.
Manligheten,	*Fylgia*,	

It is important to note in connection with the Swedish navy that Messrs. Yarrow have just constructed for that service the *Mode*, which is one of the finest examples of modern destroyer practice. Her speed places her among the fastest boats in the world, since on a three-hours' endurance trial the boat made, when fully loaded, 32.4 knots per hour. The mean air pressure in the fire rooms was 1.5 inches.

In the Norwegian navy are two ironclads, *Norga* and *Eidsvold*, each of 4,500 horsepower, built by Messrs. Armstrong, Whitworth & Co., and both these vessels have an installation of Yarrow boilers.

The Portuguese navy have also adopted this type of boiler in their first-class cruiser *Don Carlos I*, having an installation of 12,500 H.P.

ITS ADOPTION FOR SERVICE IN BRITISH CRUISERS AND BATTLESHIPS.

Having thus traced the introduction of the Yarrow boiler into the various navies, it is particularly interesting to deal more specifically with the events which induced the British Admiralty to adopt it for service in some of its cruisers and battleships.

Prior to 1900 the Belleville boiler had been almost exclusively adopted by the British Admiralty as the water-tube boiler of

approved type for cruisers and battleships. As the British experience with this steam generator became more extended, confidence in its efficiency and endurance decreased and thus led to the appointment of a committee, under Vice-Admiral Sir Compton Domville, to ascertain, practically and experimentally, the relative advantages and disadvantages of the Belleville boiler for naval purposes. The opinion of this committee as to the value of the Belleville boiler is well known, the result being that no further adoption of this type of steam generator as the approved type is contemplated in the British navy.

In another part of the precept of the Admiralty, the committee were requested to report whether any other description of boiler had sufficient advantages over the Belleville, the Niclausse or Babcock & Wilcox, as a boiler for large cruisers and battleships, to justify its installation in any of H. M. ships for trial.

In their interim report of 19th February, 1901, the committee stated that they had considered four types of straight-tube boilers, among them being the Yarrow design, and they unanimously recommended that H. M. S. *Medea* be placed at the disposal of the committee for experiment with Yarrow boilers.

To this the Admiralty readily consented, and the *Medea* was therefore fitted with eight Yarrow boilers, each of the following dimensions:

Heating surface, square feet, . . .	2,896
Grate area, square feet, ...	56
No. of tubes, ...	1,008
Outside diameter of tubes, inches, .	$1\frac{3}{4}$

Weight of each boiler with water at working level, 30 tons, each boiler indicating 1,000 H.P.

PROGRESSIVE INCREASE IN THE INSTALLATION OF YARROW BOILERS.

The installation of boilers of the Yarrow design in cruisers and battleships of Continental navies has been extensively carried on.

The following table shows the warships fitted with such boilers, the greater part of this installation having been done during the past few years:

CRUISERS AND BATTLESHIPS BUILT AND BUILDING WITH YARROW BOILERS (EXCLUDING DESTROYERS AND TORPEDO BOATS AND THE SMALLER CLASS OF VESSEL.)
JANUARY, 1903.

Type of vessel.	Built or building.	Name.	I.H.P.	Boilers.
		AUSTRIAN NAVY.		
Cruiser	Built	*Zenta*	7,800	All Yarrow.
Cruiser	Built	*Aspern*	7,800	All Yarrow.
Cruiser	Built	*Szigetvár*	7,800	All Yarrow.
Cruiser	Building	"*E*"	13,000	All Yarrow.
Ironclad	Building	"*A*"	14,000	All Yarrow.
Ironclad	Building	"*B*"	14,000	All Yarrow.
Ironclad	Building	"*C*"	14,000	All Yarrow.
		BRITISH NAVY.		
Cruiser	Building	*Hampshire*	22,000	⅘ths Yarrow.
Cruiser	Building	*Antrim*	22,000	⅕th cylindrical.
Cruiser	Building	*Amethyst*	9,800	All Yarrow.
Cruiser	Building	*Diamond*	9,800	All Yarrow.
		CHILIAN NAVY.		
		(*Built by Messrs. Armstrong, Whitworth & Co., Ltd.*)		
Ironclad	Building	*Constitucion*	12,500	All Yarrow.
		(*Built by Messrs. Vickers, Sons & Maxim, Ltd.*)		
Ironclad	Building	*Libertad*	12,500	All Yarrow.
		DUTCH NAVY.		
Cruiser	Built	*Holland*	10,000	7,500 H.P. Yarrow and 2,500 H.P. cylindrical.
Cruiser	Built	*Friesland*	10,000	
Cruiser	Built	*Zeeland*	10,000	
Cruiser	Built	*Utrecht*	10,000	All Yarrow.
Cruiser	Built	*Gelderland*	10,000	All Yarrow.
Cruiser	Built	*Noord Brabant*	10,000	All Yarrow.
Ironclad	Built	*Koningin Regentes*	6,000	All Yarrow.
Ironclad	Built	*DeRuyter*	6,000	All Yarrow.
Ironclad	Built	*Hertog Hendrik*	6,000	All Yarrow.
Ironclad	Building	Not yet named	6,000	All Yarrow.
		NORWEGIAN NAVY.		
		(*Built by Messrs. Armstrong, Whitworth & Co., Ltd.*)		
Ironclad	Built	*Norge*	4,500	All Yarrow.
Ironclad	Built	*Eidsvold*	4,500	All Yarrow.
		PORTUGUESE NAVY.		
		(*Built by Messrs. Armstrong, Whitworth & Co., Ltd.*)		
Cruiser	Built	*Don Carlos I.*	12,500	All Yarrow.
		SWEDISH NAVY.		
Cruiser	Built	*Psilander*	4,500	All Yarrow.
Cruiser	Built	Class *Uggla*	4,500	All Yarrow.
Ironclad	Built	*Dristigheten*	6,000	All Yarrow.
Ironclad	Built	*Aran*	6,000	All Yarrow.
Ironclad	Built	*Wasa*	6,000	All Yarrow.
Ironclad	Built	*Tapperheten*	7,500	All Yarrow.
Ironclad	Building	*Manligheten*	6,000	All Yarrow.
Cruiser	Building	*Fylgia*	12,000	All Yarrow.
		Total I.H.P.	308,700	

U. S. Navy.—The light-draught gunboat *Nashville* has an installation of 2,536 I.H.P. and the gunboat *Scorpion* (converted yacht) has an installation of 2,800 I.H.P.

DEVELOPMENT OF THIS TYPE OF BOILER FOR CRUISERS AND BATTLESHIPS.

The progressive increase in the size and evaporative power of the boilers of this design is graphically shown in Sketch 1. The first boilers were installed in 1887, in small second-class torpedo boats. Then larger boilers were progressively designed for the first-class torpedo boats and destroyers, concluding with those which are now being used and installed in first-class cruisers and battleships.

SPECIAL FEATURES IN THE DESIGN AND CONSTRUCTION OF THE YARROW BOILER.

The guiding principle in the design of the Yarrow boiler has always been simplicity and a strict avoidance of screwed and bolted joints. So prominently has this been kept in view that, with the exception of three ordinary manhole-plate joints, there are no steam or water fittings to make before steaming, and none to break before examination.

The Yarrow boiler is the product of the best methods of boiler making, and does not consist of a conglomeration of screwed units.

In analyzing the design and construction of this boiler it should be noted that there are neither screw plugs nor bolted plates for expanding and examining the interior of the tubes, as in some other types. In course of time leakage must necessarily take place from plates or plugs, since carelessness in making or breaking the joints will eventually cause them to leak. Every practical engineer will understand that it is impossible to keep all these joints absolutely tight, although when new there may be no leakage of either steam or water. Consequently, as regards loss of water and the size of the ship's evaporating plant, it is considered that the Yarrow boiler stands on a par with the ordinary return fire-tube type, as there is practically no leakage whatever in the boiler proper except that due to the safety valves. The

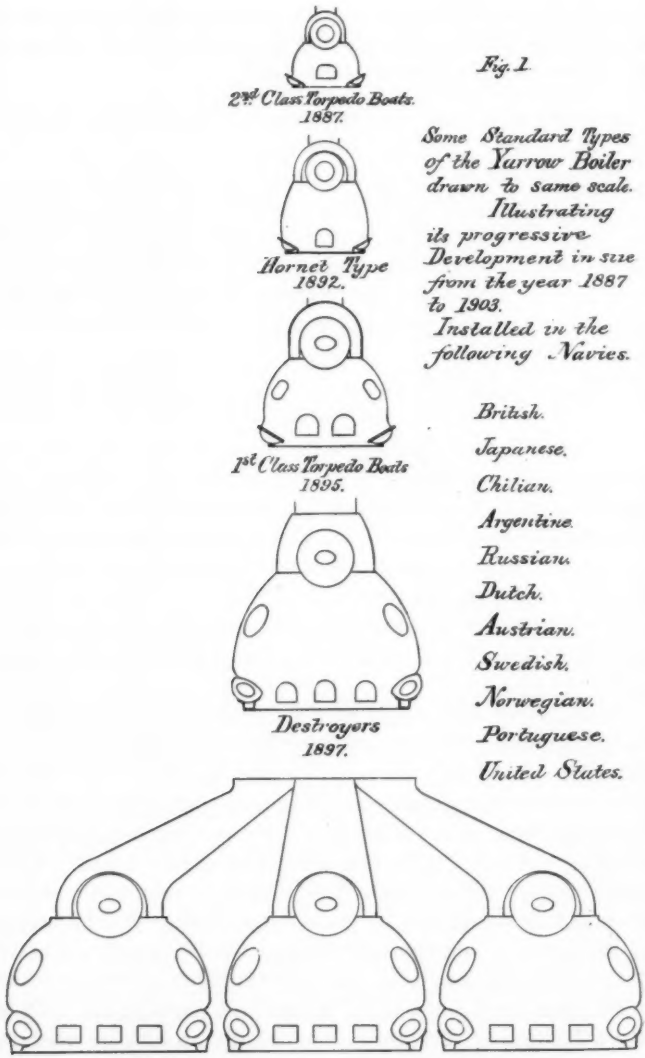


Fig. 1.

Some Standard Types of the Yarrow Boiler drawn to same scale. Illustrating its progressive Development in size from the year 1887 to 1903. Installed in the following Navies.

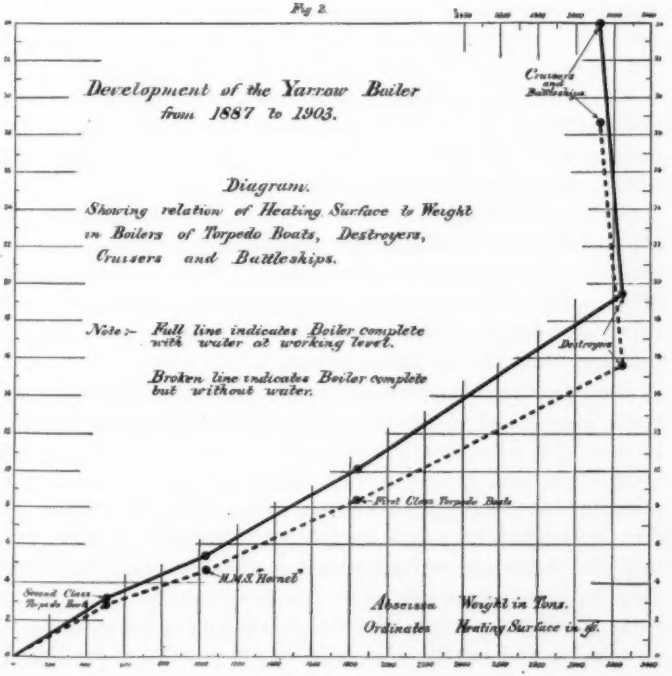
British.
Japanese.
Chilian.
Argentine.
Russian.
Dutch.
Austrian.
Swedish.
Norwegian.
Portuguese.
United States.

tendency of any boiler to leak by reason of the character of its construction is a matter of importance in naval engineering, since both the efficiency and endurance of a water-tube boiler is dependent to a marked degree upon its freedom from even small leaks.

RELATION OF WEIGHT AND HORSEPOWER.

It will be seen from Sketch 2 that the weight of boilers for torpedo boats and destroyers increases progressively with the heating surface. For cruisers and battleships, however, the weight is greatly augmented in consequence of the thicker material that must be used in securing greater durability in boilers installed in vessels that have to remain at sea for lengthened periods.

Fig 2.


Development of the Yarrow Boiler from 1887 to 1903.
Diagram.
Showing relation of Heating Surface to Weight in Boilers of Torpedo Boats, Destroyers, Cruisers and Battleships.
Note:- Full line indicates Boiler complete with water at working level.
Broken line indicates Boiler complete but without water.
Cruisers and Battleships
Destroyers
First Class Torpedo Boats
H.M.S. Hornet
Second Class Torpedo Boats
Abscissa Weight in Tons.
Ordinates Heating Surface in ft.

As regards the weight of this form of water-tube boiler, it may be stated that for a certain generation of steam, and a definite amount of heating surface, and assuming equal durability with all types, the weight is practically the same whether for the Belleville, Niclausse, Babcock & Wilcox, Thornycroft, Normand or Yarrow boiler. Where there is a marked difference in weight, investigation will show that it is due to some of the parts of the lighter boiler being thinner, and therefore less durable. It is, of course, possible, by making small drums, using thin tubes, employing light casings, and building but little brick work, to lighten up on boiler weights; but where saving has been effected in this manner, the boiler will prove satisfactory only for official trial conditions, since such construction will not give the boiler endurance, and therefore trouble will be encountered shortly after the vessel is put into service.

THE QUESTION OF CIRCULATION.

The most important feature in the design of the Yarrow boiler, and one which marks it out as distinct from all other water-tube boilers, is its straight tubes, which are placed nearly vertically. In this respect the boiler is unique.

Owing to the angle at which the tubes are placed, steam rapidly passes away to the steam drum, thereby setting up automatically a very vigorous circulation. The circulation is so efficient that boilers of this design can be severely forced in times of emergency without distressing them, a consideration that appeals to the executive officer.

It would almost appear self-evident that the circulation will be far more active in a boiler with tubes nearly vertical than it is in one with tubes nearly horizontal. Where the tube is only slightly inclined to the vertical the steam must have a freer passage to the steam chest or drum. As a result, the heating surface of the vertical tubes cannot get overheated as they may when the tubes are nearly horizontal, when the bubbles of steam do not so freely pass off. It is, therefore, only necessary in this form of boiler to provide adequate down-casts to insure complete circulation, and surely the experience and practice of the Yarrow

Company ought to show that this point has been carefully determined by both theory and practice.

CLEANING AND RENEWAL OF TUBES.

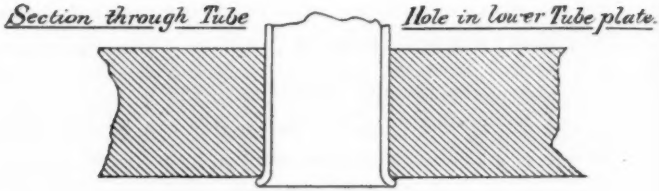
The replacing of the tubes is a matter that can be very easily carried out, and any defective tube can be removed and replaced in a few hours. It is a most important point also to be able readily to clean the outside of the tubes of all water-tube boilers. The tubes of the Yarrow can be cleaned in three directions—from the front between the tubes, from the furnace or from the casing outside. The tubes are all straight, except that occasionally the two rows next the fire are bent slightly.

The tubes, being straight, can be examined internally throughout their entire length; and this is carried out by means of a small electric light, which can be passed through the tubes and a careful examination made from the water pockets or steam chest. This electric light can be obtained from a storage cell, or from a small dynamo driven by manual power.

While the interior of the tubes can be closely examined with facility, it should be observed that any tube can be easily replaced. The boiler can thus be retubed, if desired, without removal from the vessel.

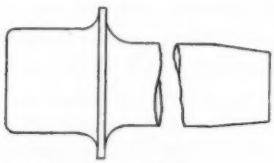
The tubes being of uniform diameter, and of simple, straight lengths, the question of spare gear or repairs is not a vexatious and serious one for the engineer. Providing they are of the longest lengths, the spare tubes can take the place of any tubes in the boiler, and it is a great point that all the Yarrow boilers in every navy have tubes of a uniform diameter, so that a supply of this size of tube in any dockyard should be common to all vessels.

These tubes are secured to the tube plates by the method which prevails in all sound boiler-making practice, *i. e.*, that of making the tube tight with an expander—a joint which does not leak if properly carried out. They are not only well expanded, but the ends projecting out of the plate for a length of about $\frac{5}{16}$ inch are then turned over to an angle of 45 degrees, as per following sketch:



This is called bell-mouthing.

The bell-mouthing is carried out by means of a tool of the following shape—



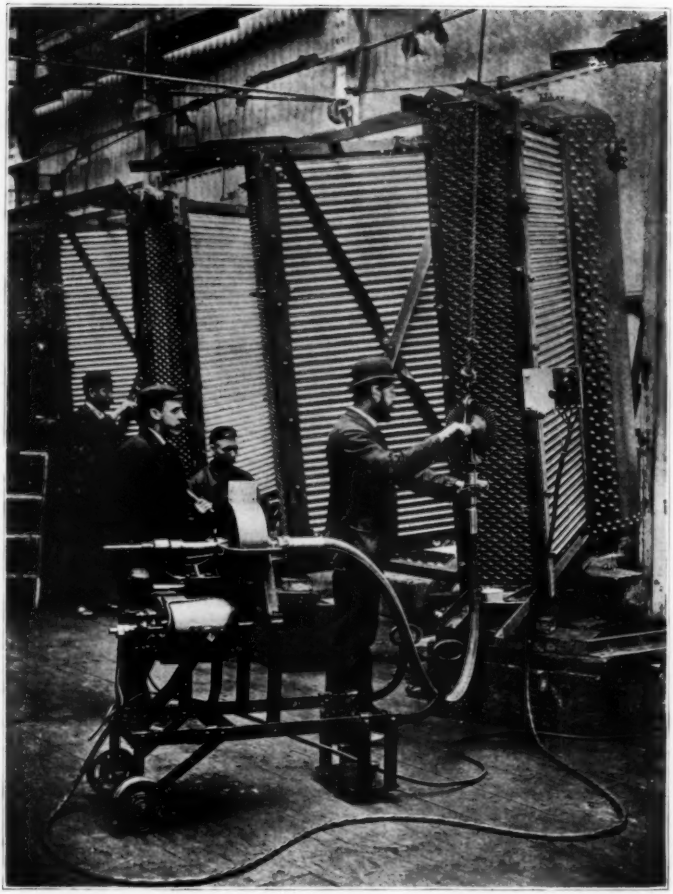
Finishing Tool.

fitted to a pneumatic hammer. It is quickly done, and should the tube split in belling it is at once rejected as being unfit for service. So great is the importance attached to the bell-mouthing that no tube is finally accepted till it has been thus treated without developing any sign of fracture.

Tubes so securely attached to the plate cannot "creep" or in any way move in the tube holes—a defect common to some types of water-tube boilers. More particularly does this occur in boilers where it is impossible to enter the water pocket to bell the tubes or to ascertain to what extent the tubes are secured to the plates.

In the construction of the Yarrow boiler the tubes are expanded and belled while every tube is open to daylight and can be subjected to the most searching examination.

The tube plates, steam-drum ends and water-pocket ends of the boilers are always pressed to shape in machined blocks and dies. This insures accuracy of form. The steam drums are



EXPANDING TUBES BY ELECTRICITY,
AS ADOPTED BY MESSRS. YARROW & CO., LTD.
POPLAR, LONDON, ENGLAND.

bored out to receive the ends, and then the ends are turned in a lathe, and everything is thus made a perfect engineering fit. By these methods water tightness of work is insured, and the amount of caulking of seams is reduced to a minimum.

When the Yarrow boiler was first made, a commercial expander was used, although, of course, individual manufacturers and chief engineers of ships preferred to use expanders of special design. Realizing the importance of properly expanding the tubes in the boilers of torpedo boats and destroyers, Messrs. Yarrow and Company greatly modified one of the leading forms of expanders and made the tool specially suited for this class of work. Commercial reasons, as well as the urgent demand for reliable, uniform and rapid expanding of the tubes, compelled attention to be given to the question of expanding tubes by mechanical means. Messrs. Yarrow & Co. were among the first to attach the expander, with a flexible shaft, to an electric motor. Owing, however, to the difficulty of reversing the motor shaft, they, at a later stage, abandoned electrical driving in favor of compressed air, the expander being rotated by the pneumatic drilling machines that are so well known and which owe their origin to American ingenuity and enterprise.

ADVANTAGES OF AN UNIFORM SIZE OF BOILER TUBES.

The advantages of adopting only one size of tube for boilers of large vessels of the cruiser and battleship class in any given navy is evident. Where more than one size of tube is used there is always liability of the wrong size being sent when there is a shortage of any one class. Then, again, there is the annoyance and evil of handling different size tubes in making repairs. Simplicity of spare gear and ready interchangeability of parts on board the warship constitutes a great military advantage. It is likewise desirable to have an interchangeability in the tools to be used in retubing, and this can be easily done if all the tubes are of uniform size. The rapidity of retubing boilers would also be increased by having uniformity, since it ought not to be a difficult matter to train even a fireman to expand one particular size of tubes.

CASINGS AND UPTAKES.

In the construction of the Yarrow boiler the question of encasing has been carefully gone into. The outcome of a wide experience has established the fact that the best non-conducting casing is millboard asbestos enclosed between two carefully leveled thin steel sheets.

Great importance is attached to all junctions of the casings being made perfectly airtight.

So efficient has this casing proved itself when properly fitted, that notwithstanding the fierce heat in the combustion chamber the paint does not blister upon the exterior of the shell. The construction of the casing is sometimes not given the attention it deserves, but upon its being carefully fitted depends to a very great measure its durability and efficiency.

SUPERIOR HEAT DISTRIBUTION IN A SEMI-VERTICAL TUBE BOILER.

Considerable evaporation is effected in every boiler by radiated heat. For example, in the fire box of a locomotive it is well known that the fire box itself, through radiated heat, generates a considerable amount of steam. In general, it is advisable wherever possible to secure evaporation through radiation, for where the hot gases suddenly impinge upon the tubes there is a tendency to subject the tubes to undue strain. In comparison with boilers having tubes nearly horizontal it will be seen that in the Yarrow boiler there is about double the heating surface for the transmittal of heat by radiation. The advantage of heating tubes from radiated surface is more clearly shown in the burning of liquid fuel, where brick walls and arches have been specially built for distributing the heat transmitted from the impinging of the heated gases.

THE MOST EFFICIENT MANNER OF OPERATING WATER-TUBE BOILERS. RELATIVE MERITS OF SCOTCH AND WATER-TUBE DESIGNS.

In connection with the subject of the relative merits of Scotch and water-tube boilers, and the relative advantages of small and large-sized tubes for steam generators of cruisers and battleships, it should be stated that the cause in many cases for the excessive coal consumption of water-tube boilers is probably due to the

fact that too little coal has been consumed per square foot of grate surface. The elasticity of construction of the water-tube boiler makes it fully able to stand forcing, and therefore efficiency and economy is more likely to be obtained by burning forty pounds of coal per square foot of grate in a few boilers rather than by using twenty or thirty pounds per foot on a much larger grate area. It may be easier work for the stokers to employ extra boilers, but where this is done it is extremely likely that uniform firing will not be carried on, and that holes in the fires will be found through which the cold air will enter, thus producing incomplete combustion and causing undue loss of heat.

It must be remembered that the furnaces in an ordinary Scotch boiler have each only about twenty-one square feet of grate surface, and therefore it is not a difficult matter for the ordinary fireman to keep the grate completely covered. In many of the water-tube boilers there is between fifty and sixty square feet of grate surface which is comprised within a single furnace. It not only requires some skill to fire a grate that is seven or eight feet long, but it necessitates conscientious work. The lack of efficiency in many cases of water-tube boilers has been too often due to the inefficient and careless way in which the coal has been thrown upon this grate. The need of a training school for stokers for such class of boilers is very urgent, and until stoking in the water-tube boilers is as efficient as is carried on in the Scotch design loss of efficiency must be the result.

The reason for burning a large amount of coal rather than a small quantity per square foot of grate is because combustion is more complete where rapid combustion is effected. The temperature of the furnace is also maintained at a higher point by forcing the fires to some extent, thus completing combustion before the gases pass through the tubes. In some designs of water-tube boilers a slow combustion may be necessary, because in such designs of boiler if the rate of combustion is very high the heat absorbed by the lower row of horizontal tubes may be too great for the generated steam to be gotten rid of; consequently the tubes get overheated and give trouble. In the case of the Yarrow boiler, owing to the very rapid circulation, it

ought to be impossible to overheat any of the tubes if there is water in the boiler, or if the tubes themselves are not choked by obstruction.

If systematic comparative experiments were made with practically similar ships, possessing water-tube and Scotch boilers, the manifest superiority of the water-tube type would be conclusively shown, for naval purposes particularly, when a large amount of coal was burnt per square foot of grate surface in both types of boilers. Such experiments would undoubtedly establish the fact that there was great loss of efficiency in water-tube boilers by burning too little coal, and that the flexibility and elasticity of the water-tube design particularly adapts it for severe work.

The stokers of the several navies have had years of excellent training and experience in familiarizing themselves with the best manner of operating Scotch boilers, while it must be remembered that as a rule untrained and unskilled men are now often necessarily assigned to the task of looking out for the water-tube type. It will be but a few years before the water-tube boiler will be on an equality with the Scotch boiler in this respect, and then there will be secured increased economy, efficiency and endurance from the water-tube type.

ECONOMY IN COAL CONSUMPTION.

The coal consumption of Yarrow boilers is highly satisfactory when compared with those of any other design. Particularly is there economical consumption when they have been in use for some time. The reason for efficient economy is because the combustion chamber is large, thus permitting an early and proper mixture of air with the gaseous carbon.

The completion of combustion before the gases pass amongst the tubes is a matter of vital importance, because the impinging of only partly consumed gases has a tendency to put out any flame and thus prevent complete combustion.

Combustion is more likely to be impaired from the above cause in boilers fitted with horizontal than with vertical tubes. Where combustion has been checked in this manner, it is often

completed again at the base of the stack, and this accounts for flaming gases issuing from the funnel of some steamers. Such a sheet of flame not only shows loss of economy, but is a matter of serious military disadvantage to the war vessel. On a dark night it might make known to the enemy the presence of an opposing fleet.

EXPERIENCE HAS PROVED THE MERITS OF DESIGN.

While the Yarrow boiler has been developing in size, the general design has consistently remained the same since its inception. Its success as a large-type boiler in ocean-going steamers has justified the adhesion to the main principles laid down in the first Yarrow boiler ever constructed.

It may safely be affirmed that no water-tube boiler of its age has altered less in design since its introduction than the Yarrow boiler, and that no other water-tube boiler requires less complicated machinery for its construction. Its extended installation on board cruisers and battleships can, therefore, be confidently expected.

THE EDUCATION OF NAVAL ENGINEERS FOR FUTURE NEEDS.

By Passed Assistant Engineer Charles H. Manning, U. S. N. (Retired), Member.

With the abolition, by the Personnel Bill, of the Engineer Corps, the distinct training of young officers in the Navy along engineering lines came to an end. Since the enactment of that law the need in the Service of engineering knowledge and experience has increased several fold, while the supply on the whole has diminished.

While it is true that 150 warrant machinists have been taken into the Service, nine-tenths of these men were formerly in the Navy. In an official report Rear Admiral Melville thus speaks of these men: "It should be remembered that when the personnel law went into effect there were in the engine rooms of our warships, in addition to an efficient commissioned Engineer Corps, an exceedingly well-trained and intelligent complement of skilled machinists. These men had been drilled by engineer graduates of the Naval Academy, and a state of efficiency had been reached in our engine rooms that had not been surpassed by that of any other service. In the competitive examination that was held for the position of warrant machinist just after the close of the Spanish-American war, the character and ability of the machinists of the Navy were put to an actual test. Almost without exception, the successful competitors came from the naval ships. These men were not only well acquainted with the machinery of our war vessels, but they had been taught by example and precept the necessity of keeping every machine in a high state of efficiency. They had witnessed the courage and devotion to duty upon the part of the engineer commissioned officers, and thus had not only been made to implicitly obey, but they had been taught the necessity of efficiently directing

affairs, so that when promotion was tendered such enlisted men they were competent to perform higher duties that could not be safely entrusted to those without naval experience."

The net gain, therefore, to the Navy, as regards engineering personnel, by the appointment of warrant machinists has been more apparent than real.

In speaking of existing engineering conditions, less than a year ago Admiral Melville, in his annual report, stated: "It has been because the belief prevails in some quarters that the experience of a single cruise in the engine room will fit junior officers for the duty of a chief engineer, and that the practical duties at sea of the old engineer officers can be taken up by the warrant machinists, that there has been continued retrogression in the engineering efficiency of the Navy." * * *

"In the vessels of the merchant marine there are as many engineers as deck officers, and surely a proportion of four deck officers to one engine-room officer should be ample, even in a modern navy. Excluding the marine officers who have charge of the secondary batteries on some of the large ships, there are 1,856 warrant officers, midshipmen and commissioned officers in the U. S. Navy. Allowing one-fifth of this total number for engineering duty ashore and afloat, there should be at least 370 assigned to this work instead of the 300 who are now doing the duty. The request of the Bureau that such a proportion should be detailed for engineering work cannot be regarded as excessive."

A careful reading of the reports of all the Bureau chiefs, and of the special reports sent by the Navy Department to Congress, practically shows that there is no denial of the declaration that a deficiency exists as regards the number of officers doing engineering duty in the Navy. Some definite plan to supply the deficiency must soon be adopted, or the outbreak of war would find the ships very deficient in one of their most vital departments. One is but simply stating an axiom when one maintains that a modern battleship in action might as well be left without competent officers in charge of her battery as without competent engineer officers in her engine rooms.

The "personal factor" is as necessary to meet emergencies below the protective deck as in the conning tower or turret, and the final result of the battle will depend as much on everything in the engine room being at its very best as that the battery should be in the highest possible state of efficiency. Time and experience will show that efficiency in the engine room will not be secured with less training, less care, less mentality and less executive work than efficiency in the redoubt and the turret. It is because this question is the most serious one of naval administration today that the British Admiralty has given this subject far greater consideration than it has devoted to any other matter relating to the Navy.

Sixty years ago, just after steam was introduced into the Navy, the engineers were principally recruited from the polytechnic schools of that day. In those days the machine shop was the technical school, for by reason of the fact that the artisans had to serve a long period as apprentices, and as advancement did not come unless the apprentice studied after working hours, there entered the Navy as engineers many men who had not only manual skill, but who were close students of technical affairs. The training of that day made it necessary for young men to acquire a habit of concentration and application, and thus there came into the Navy young officers who were receptive for information. Our advance as regards naval engineering matters was as well recognized from 1855 to 1865 as it was from 1885 to 1895. From 1840 to 1865 there also entered the Engineer Corps of the Navy graduates of the Troy Polytechnic Institute, Union College, Lawrence Scientific School of Harvard University, and also from other institutions which gave a scientific course of instruction.

This method of recruiting carried us safely through the Civil War. It was found, however, that the needs of the future demanded systematic and continuous training of naval-engineer officers. It was noticeable that when recruiting of the Engineer Corps was dependent upon voluntary applications that there was too much uncertainty both as regards quantity and quality as to what the Navy would receive. There would enter one year a

remarkably strong class, and then might follow two or three classes of mediocre ability.

It was Von Moltke who declared, about thirty-five years ago, that war was a business, and when this fact became to be appreciated in its fullness by our naval administrators it was realized that the time had now come when the Navy Department should make provision whereby a definite quantity and quality of engineer officers should be trained each year for naval needs. In pursuance of this policy a detail of Third Assistant Engineers, who had graduated from technical schools, was sent to the Naval Academy for two years' training, and this body of young men comprised the first class of cadet engineers who graduated from that institution. This method was productive of good results, for it turned out some very able men, one of whom, Rear Admiral C. W. Rae, U. S. N., is now Engineer-in-Chief of the Navy.

About the same time the original Cadet Engineer law was enacted; but, for reasons useless to discuss now, it brought in but two recruits in 1866. It was not until September, 1871, that a class of 17 cadet engineers entered the Annapolis Academy for a two-years' course. In 1872 and 1873 there entered the Naval Academy 25 cadet engineers each year. Then the law was changed, extending the course for four years. It may be incidentally stated that the average age of the cadet engineers entering in 1871, '72 and '73 was about nineteen years, and that practically all were graduates of high schools, and that several had completed a collegiate course.

During the ten years of the existence of the Cadet Engineer course there was probably added to the Service the bulk of its present engineering talent, not to mention the many brilliant men who have in the meantime resigned from the Service.

From the repeal of the cadet engineer law until the abolition of the Engineer Corps there were only practically enough admissions to the Corps to keep it up to the strength provided for by law. There were some excellent men who entered the Engineer Corps between 1885 and 1898, but in general the same average material could not be secured from Congressional appointments

as when the cadet engineers entered after competition. This brief review of the history of the Engineer Corps is exceedingly important in that it suggests that all the cadets who enter both the Naval and Military Academies should do so after competitive appointment.

The strength of the system of having a separate class of cadet engineers did not come from the fact of the cadets doing distinct engineering work, but from the manner in which the engineers were recruited. The needs of the future can be best met by educating all alike at the Academy, and then giving post-graduate work along special lines to those who have marked aptitude for technical duties.

The method of selecting cadets and the requirements for their admission into the Academy should be radically changed. All appointments should be competitive and entirely free from political influence. In his able paper, "How Best to Meet the Navy's Greatest Need"—and this need is the proper supply and training of officers and men—Lieutenant J. S. McKean, U. S. N., thus writes of this proposition: "I believe that the system (of examining cadets) should be extended so that an examination should be held in at least every State in which there is a vacancy, and not only held by the Civil Service Board, but Civil Service rules apply. The highest among the competitors should get the appointment, not one who can simply pass a prescribed examination, and who is selected by the appointing officer for political, family or social reasons."

The superior scholarship of the cadet engineers during the time they were at the Academy was due to the manner in which they were selected. Hundreds would apply to the Secretary of the Navy for permits to appear before the Academic Board at Annapolis. A hundred or more of the most promising would be granted such permission. From those who took the examination twenty-five of the best were selected, and the inevitable result was that the average standard of scholarship was higher than that of the young men who came in by Congressional appointment. The official records of the Academy will show whether or not this statement can be substantiated.

The next step in securing desirable cadets should be in the direction of increasing the standard of admission to the Naval Academy to that exacted by the first-class technical schools of the country. Every graduate of a high school who was qualified to pass the required physical examination might thus be led to believe that there existed an opportunity to seek a career in the Navy. The work and influence of the Naval Academy would thus be greatly extended. It would become a greater educational force than it has been in the past, and it would be brought much nearer the hearts of the people. Particularly would the following words of President Tucker, of Dartmouth, in regard to the work of the American college apply to the Naval Academy: "The American college or university stands for social advancement, as well as for intellectual discipline. The university is that gateway through which democracy passes to the refinement of its strength." If the thousand cadets whom the rehabilitated Naval Academy is to contain should center only after competitive examination, and if the standard of admission should be raised, then the Annapolis institution would not only do much to make the democracy of the nation rise to the refinement of its strength, but there would be secured for the day of battle a complement of officers whose peers could not be found in any naval service.

The third change that should be made should be in the direction of strengthening the Academy course along engineering lines. As the entering cadets would have a more entended knowledge of English studies, modern languages, physics and mathematics than is now possessed upon admission, there would be more time permissible for the acquisition of engineering principles and technical experience.

The naval career today is a technical one, and those having a distaste for mechanical affairs had better look elsewhere for a career, for the principal work at least of the junior naval officers will hereafter relate to electrical, pneumatic, hydraulic and steam appliances. There will be scant time for poetry, and mooning, and vain regrets for the disappearance of the braces and the halliards.

Every cadet should spend at least one summer at one of the large navy yards at actual work, and "bell hours" should be spent by them in the engineering, construction and ordnance shops. While under instruction at a navy yard the cadets should be comfortably quartered in barracks on shore, and not hung up in "dream sacks" on some small tug or sailing brig. A second summer should be devoted to visiting the leading ship yards, gun foundries and manufacturing establishments. The cadets should be made to take careful and extensive notes, for this would teach them both how to observe and how to listen—not to mention the skill which they would acquire in sketching.

The practical side of education should be given much weight, because for those who possess application and who have learned how to study there will be plenty of science and theory available in the books that will be found in the various ships' libraries, or which can be purchased at comparatively slight cost.

Post-graduate work is essential in every professional career, and such work should be done at the Naval Academy. But every graduate of the Academy is neither entitled to receive nor would many care for such advanced instruction. Therefore, the study of abstruse text books and the effort to make designers and original thinkers of all graduates should not be attempted.

The belief thus exists in many quarters that the curriculum of the Naval Academy is not too difficult but too extended, and that some of the studies now carried on should be reserved for post-graduate work. It is certain that the engineering demands of the future will require post-graduate work upon the part of those to whom must be entrusted important technical duties.

Fortunately for our naval needs, within a few years the projected naval-engineering laboratory—that will stand as Admiral Melville's best monument—will be in operation, and the influence of this experimental station is going to be very far-reaching in showing the cadets the scope and character of engineering work. With such a laboratory in existence it will be an easy matter for the Academic Board to carry on post-graduate work

of the most advanced nature. This laboratory is going to make for future naval engineering efficiency, while at the same time it should do much in the solution of many unsolved problems that now confront naval architects and marine engineers.

After four years of undergraduate work at the Academy all the cadets should be sent to sea. When one considers the standard of admission to the Naval Academy as compared with that of the leading scientific schools, it can be well appreciated that a four years' course is not any too long. It is during the last year when the best professional work must be done, and it is doubtful if either the Service or the individual cadet is benefited by interfering with the Academic curriculum by graduating the cadets in the middle of the term. It is much easier to demoralize educational work than to improve its character, and surely if the cadet becomes of the opinion that he is to graduate in advance of the scheduled time, he is not likely to study as hard the last year as he would otherwise be inclined.

Half of the cadet graduates after being ordered to sea-going vessels should be assigned to duty on deck for one year, and half to duty in the engine room. Those doing duty on deck or in the engine room should alternate at the end of a year in their work, but there should be a transfer to another ship when the change of duty takes place. Alternate watches for brief periods either on deck or in the engine room is the worst mistake possible, as it will destroy interest in both duties, and a man without interest in his work, be he preacher or pirate, will never accomplish his purpose.

At the end of the two years' cruise the policy of specializing might be tried. At the end of that period it should be manifest whether the inclinations of some officers are in an engineering direction. Those whose inclinations are strongly in that direction might be assigned to permanent engineering work, but it should be fully understood that junior officers seeking such duty should have all the pay, prestige and emoluments of those continuing line work. If in any way there is discrimination against engineering duties, the inevitable result will be engineering retrogression.

No matter how attractive the curriculum, nor how extended may be the training given at the Naval Academy, or anywhere else, there can be no success in developing naval engineers unless love and pride of the profession exist in the individual. The expense incurred in rehabilitating the Naval Academy can thus best be justified by making the institution preëminently strong along naval-engineering lines. If the training is otherwise, the question may well be asked: Will the new marble palaces at Annapolis, in which the future cadets are to be trained, and which are classic in outline, make the cadet more contented with his lot, and better fitted for professional work when he goes to sea and finds that engineering, and not classic, conditions confront him?

OFFICIAL TRIALS OF THE SUBMARINE BOATS *PLUNGER, PORPOISE* AND *SHARK.*

By Lieutenant Daniel S. Mahony, U. S. N., Member.

The official trials of the submarine boats *Porpoise*, *Plunger* and *Shark* took place in Peconic Bay, Long Island, May 13–21, 1903. These three boats being sister craft of the *Adder* and *Moccasin*, described in the January number of the Journal, and of the *Grampus* and *Pike*, described in the May issue, need no further description here.

The standardization trials took place over a half-mile course. As the boats entered and left the course flags were dropped by two observers on shore, the times being taken by another observer on a tug following the submarine. This procedure was necessary, as one end of the shore base line was not visible at the other end. The observers on the boats could not see when the boats entered and left the course, so the revolutions were taken for a period of three minutes while the boat was on the course, and the average revolutions thus obtained per minute were considered the average for the run. Each boat was standardized in three conditions; light, awash and submerged. After standardization each boat had to make the following runs:

1. Ten miles in the light condition, under the gasoline engine, at a speed of at least eight knots.

2. Ten miles in the awash condition, under the gasoline engine, at a speed of at least seven knots.

3. Two miles submerged, under the electric motor, firing a torpedo at a target at the end of the run.

One boat was required to make a run of twelve hours under the gasoline engine at a speed of at least eight knots. This being accomplished successfully, all the boats were to be considered satisfactory in this respect. One boat was to make a run

submerged under the motor for three hours at a speed of at least seven knots. This being accomplished, all the boats were to be considered satisfactory in this respect.

In order to give the readers of the JOURNAL absolute data as to the various runs, the writer has been kindly permitted by Naval Constructor J. J. Woodward, U. S. N., to use the notes that were carefully made by him on the trials. A copy of Mr. Woodward's notes is given for each submerged run, as it appears to the writer that the actual work done by the boats cannot be set forth more clearly than in these notes.

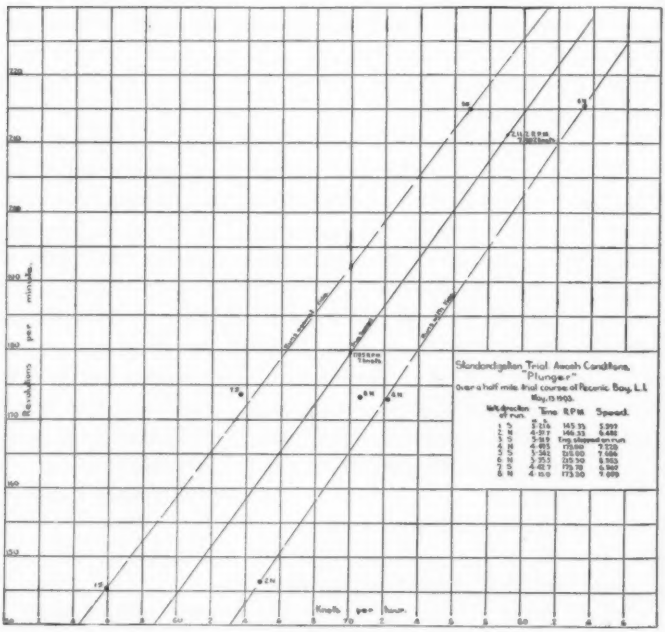


FIG. 1.—U. S. S. "PLUNGER," SUBMARINE BOAT. STANDARDIZATION TRIAL, AWASH, MAY 13, 1903.

SUBMARINE TORPEDO-BOAT *PLUNGER*.

On the afternoon of May 13th the *Plunger* was brought to the course and an attempt made to standardize the boat submerged.

It was found impossible to close the hatch over the conning tower, the rubber gasket being too thick on the side next the hinge, thus keeping the door from closing tight on the side opposite. The runs submerged had to be abandoned, and the *Plunger* was standardized in the awash condition, running with the gasoline engine. Runs with and against the tide were made at three speeds, as follows:

1. Under two cylinders of the gosoline engine, with delayed ignition.

2. Under two cylinders of the gasoline engine, with early ignition.

3. Under three cylinders of the gasoline engine, with early ignition.

The boat finished the trials successfully. On one double run a mean speed of 8.02 knots having been made. The revolution speed curve for this trial is indicated in Fig. 1.

On the afternoon of May 14th the *Plunger* successfully finished her ten-mile speed trial in the awash condition, using three cylinders of the gasoline engine. The trial lasted an hour and a half, the mean speed being 7.9 knots per hour, and the average revolutions per minute 211.12.

The *Plunger* was standardized submerged on the afternoon of May 15th. A flag on a tall mast near the bow of the boat could be seen by the observers on shore while the boat was submerged. It was intended to run at three speeds, but, the "A" frame shaft bearing running hot, the trial was abandoned after two double runs. As over seven knots were made on the last double run, no further trials were made submerged. It was found that 174.4 revolutions per minute were necessary for a speed of seven knots.

NOTES TAKEN DURING THE STANDARDIZATION TRIAL, SUBMERGED, MAY 15, 1903.

TIME. h. m. s.	
2·22·30	Began to prepare for running boat submerged.
2·25·00	Turned air from supply flask on the air system of piping for use in emptying tanks.
2·27·30	Blew small quantity of water out of the auxiliary tank.
2·29·50	Filled torpedo tube.

h. m. s.

2·35·00 Took readings of storage batteries on open circuit, as follows: Forward battery, 61 volts; after battery, 61 volts. 10 ampères were being used for lighting the vessel.

2·37·50 Order given to fill main tank.

2·40.50 Shut off main tank.

2·42·50 Order given to go ahead filling main tank.

2·43·30 Shut off main tank.

2·44·00 Shifted lead ballast near torpedo tube.

2·49·30 Order given to fill main tank.

2·50·10 Main tank reported full.

2·50·30 Blew out after trimming tank.

2·51·40 After trimming tank reported empty.

2·52·05 Order given to fill auxiliary tank.

2·53·40 Shut off auxiliary tank.

2·53·40 Order given to fill forward trimming tank.

2·58·40 Forward trimming tank reported full.

2·58·45 Order given to fill both compensating tanks under torpedo tube.

3·02·40 Compensating tanks under tube reported full.

3·03·00 Order given to fill auxiliary tank.

3·04·10 Shut off auxiliary tank.

3·04·40 Order given to fill gasoline tank (with water to gain additional weight by head).

3·06·15 Shut off gasoline tank.

3·07·30 Order given to fill auxiliary tank very slowly.

3·07·52 Shut off auxiliary tank.

3·08·12 Started ahead on motor running on surface; revolutions per minute, 174.

FIRST RUN.

3·10·00 Order given to dive going south on first run of submerged standardization. Approximate conditions of steady motion: trim, 3 to 4 degrees by head; depth, 8 to 10 feet; diving rudder, 3 degrees dive; revolutions per minute, 155.33.

3·17·40 Rose to the surface.

h. m. s.

3·18·00 Stopped motor and set up gland on shaft of steering wheel going to the upper deck where there had been a slight leak; also the cover of the bottom of the deck ventilator belonging to the blower near the gasoline engine. At rest on the surface, the trim was 1½ degrees by the head, the depth gauge read 4 feet.

3·25·00 Started ahead on the motor, revolutions per minute on the surface being 160.

SECOND RUN.

3·28·00 Order given to dive.

3·29·00 Reached depth of 10 feet by gauge. Approximate conditions of steady motion: trim, 3 degrees by head; angle of diving rudder, 3 degrees dive; depth by gauge, 10½ feet; revolutions per minute, 152.66. Other observations for approximate condition of steady motion during this run gave as follows: trim, 4 degrees by head; depth by gauge 9½ feet; angle of diving rudder, 3 degrees dive. Another set of observations gave: trim of vessel, 4 degrees by head; depth by gauge, 9½ feet; angle of diving rudder, 4 degrees dive.

THIRD RUN.

While running on surface, turning, the approximate conditions of steady motion were—trim of vessel, 2 degrees by head; depth by gauge, 2½ feet; angle of diving rudder, 6 degrees dive.

3·42·00 Order given to dive. The maximum angle of trim while making this dive was 9 degrees. The average depth during this run was 10 to 12 feet; revolutions per minute, 183.66.

3·48·00 Rose to the surface.

FOURTH RUN.

3·57·00 Order given to dive. Average depth maintained during this run, 10 feet. A slight amount of smoke was noticeable in the after part of the boat, apparently due

h. m. s.

to the heating of the "A" frame bearing of the main shaft. Approximate conditions of steady motion noted during this run—trim of vessel, 3 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 5 degrees dive; revolutions per minute, 179. Another observation for conditions of steady motion made during this run gave—trim of vessel, 2 degrees by head; depth by gauge, 10½ feet; angle of diving rudder, 5 degrees dive.

4·03·00 Rose to the surface. Maximum angle of trim of vessel while rising, 8 degrees by stern.

4·04·00 Order given to blow out midship tank.

4·05·00 Stopped the motor.

4·07·15 Order given to blow out main tank.

4·11·00 Shut off main tank.

4·12·00 Order given to blow out auxiliary tank.

4·12·30 Opened the door in top of conning tower.

4·14·00 Shut off auxiliary tank.

The bearing that heated is just forward of the large gear wheel through which the motor drives the propeller shaft. The couplings in the shaft brought a thrust on this wheel, forcing it against the end of the bearing, heating the latter.

On May 16th the *Plunger* was standardized in the light condition, runs being made at three speeds, using two, three and four cylinders of the gasoline engine, with early ignition. A mean speed of 8.876 knots was made during one double run, and 224.7 revolutions per minute were found necessary for eight knots. On the same day the *Plunger* ran successfully her ten-mile speed trial in light condition, the run lasting an hour and fifteen minutes. The mean revolutions per minute were 237.6, the mean speed of the vessel 8.725 knots, and the total distance run 10.91 miles.

On May 19th the *Plunger* ran two miles submerged and fired her torpedo, which hit the target exactly. The weather was calm and the water smooth. The mean revolutions per minute were 183.24, the mean speed of vessel 7.32 knots, and the total distance run 2.14 knots.

NOTES TAKEN DURING THE TORPEDO TRIAL, MAY 19, 1903.

TIME. h. m. s.	
9·16·00	Closed conning-tower top.
9·18·00	Order given to fill main tank.
9·18·30	Started ahead on electric motor.
9·18·35	Stopped electric motor.
9·19·30	Shut off main tank.
9·19·45	Opened conning-tower top.
9·20·00	Took voltage of batteries on open circuit, which gave the following results: Forward batteries, 61 volts; after batteries, 60½ volts.
9·21·50	Closed conning-tower top.
9·22·00	Order given to fill main tank.
9·24·50	Shut off main tank.
9·26·30	Order given to fill torpedo compensating tanks.
9·27·00	Order given to fill forward trimming tank.
9·28·10	Started ahead on propelling motor.
9·28·20	Stopped motor.
9·30·30	Shut off trimming tank.
9·30·50	Order given to fill main tank.
9·31·45	Shut off main tank.
9·31·50	Order given to fill auxiliary tank.
9·33·20	Shut off auxiliary tank.
9·33·25	Order given to resume filling forward trimming tank.
9·34·10	Shut off forward trimming tank.
9·34·20	Order given to fill auxiliary tank.
9·34·50	Shut off auxiliary tank.
9·35·10	Fill forward trimming tank.
9·36·35	Shut off forward trimming tank.
9·36·45	Fill auxiliary tank.
9·37·05	Shut off auxiliary tank.
9·37·40	Order given to go ahead very slowly filling auxiliary tank.
9·39·34	Shut off auxiliary tank, and everything.
9·39·50	Started ahead on electric motor. Conditions of steady running at surface: trim of vessel, 3 degrees by stern; depth gauge read 3½ feet; angle of diving rudder, 3 degrees dive.

h. m. s.

9·43·40 Stopped motor while schooner was towed off the course.

9·45·35 Started ahead on motor. Conditions of steady running on surface: trim of vessel, 2½ degrees by stern; depth gauge read 2½ feet; angle of diving rudder, 3 degrees dive.

9·51·10 Order given to dive.

9·52·10 Speed of motor increased to full speed for trial, the beginning of the run, including the act of diving, having been made at reduced speed, after allowing a brief interval for boat to gather way.

9·54·25 Conditions of steady motion: trim of vessel, 3 degrees by head; depth gauge read 10 feet; angle of diving rudder, 4 degrees dive.

10·00·00 For the past 8 minutes the depth was maintained within six inches of 10 feet by gauge. At this time noticed slight smoke in after part of vessel.

10·02·40 Depth of gauge, 8 feet. Vessel turning underway; moderate fluctuations in depth.

10·03.45 Depth by gauge, 12 feet.

10·04·30 Depth by gauge, 10 feet. Still turning underway. Approximate steady motion: trim of vessel, 4½ degrees by head; depth by gauge, 10½ feet; angle of diving rudder, 2 degrees dive.

10·05·35 Rose to surface.

10·06·10 Order given to dive. Maximum angle of trim of vessel while diving, 12 degrees.

10·07·12 Rose to surface.

10·07·45 Started to dive.

10·07·55 Depth by gauge, 10 feet.

10·08·15 Order given, "Get ready to fire torpedo."

10·08·30 Depth by gauge, 15 feet.

10·09·00 Depth gauge read 7½ feet.

10·09·10 Order given to "Stand by to fire."

10·09·20 Rose to surface.

10·09·40 "Fire."

10·09·50 Order given to slow down motor.

h. m. s.
10 10 10 Blow out midship tank.
10 10 40 Shut off midship tank.
10 10 50 Order given to blow out main tank.
10 11 00 Stopped motor. Smoke noticed in the after part of the boat, appearing to come from the armature. Bearing of the main shaft over the motor had heated slightly.

SUBMARINE TORPEDO-BOAT *PORPOISE*.

The *Porpoise* was standardized awash on May 14th. The engine did not work well on the first two runs, coming to a dead stop on the second run. The water in the gasoline tank was drawn along with the gasoline to the engine, stopping it. After letting water into the tank until the gasoline filled the top of it, and using the upper suction, the trial was continued. For a speed of seven knots, 208.6 revolutions per minute were found necessary.

After the runs awash, the *Porpoise* was standardized in the light condition. The trial was a failure, the boat not making a speed of eight knots. Water mixed with the gasoline was the cause of the failure. This trial was repeated on the morning of May 15th, and finished successfully. For a speed of eight knots 238.8 revolutions per minute were found necessary. The maximum speed made on a double run was 8.4125 knots.

After the standardization runs in light condition, the *Porpoise* made her ten-mile run light, the run lasting one hour and twenty minutes. The mean speed was 8.834 knots, the mean revolutions 247.9, and the total distance run, 11.18 miles.

On May 16th the *Porpoise* was standardized submerged. The sea was smooth and the breeze light. For a speed of seven knots 183.7 revolutions were found necessary, and 7.125 knots were made on one double run. It was intended to conduct the two-mile submerged run and torpedo firing immediately after the standardizing runs, but, the diving rudder working very stiff, these tests had to be abandoned.

SUBMARINE TORPEDO-BOAT *PORPOISE.*

Notes taken during the submerged standardization trial, May 16, 1903.

TIME.
h. m. s.

1·50·00 Began preparing to put boat in diving condition, sending below ventilator, deck steering wheels, etc.

1·53·20 Started blowing out auxiliary tank.

1·54·30 Shut off auxiliary tank.

1·55·00 Order given to fill midship tank.

1·57·00 Order given to fill torpedo tube.

2·19·00 Took the voltage of batteries on open circuit, which was as follows: After battery, 61¼ volts; forward battery, 61¼ volts.

2·20·10 Closed conning-tower top.

2·22·40 Order given to fill main tank.

2·24·30 Main tank reported full.

2·24·45 Order given to fill auxiliary tank.

2·29·40 Shut off auxiliary tank.

2·29·50 Started ahead on motor.

FIRST RUN.

2·32·30 Order given to dive. Approximate conditions of steady motion: trim of vessel, 2 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 3 degrees dive; revolutions per minute, 153.

2·40·30 Order given to rise. Maximum angle of trim while rising to surface, 5 degrees by stern.

While running on surface conditions steady motion noticed; trim of vessel, 1 degree by stern; depth by gauge, 1 foot; diving rudder, 5 degrees rise.

SECOND RUN.

2·46·30 Order given to dive. Approximate maximum angle of trim while diving, 9½ degrees by head. Approximate conditions of steady motion: trim of vessel, 1 degree by head; depth by gauge, 10 feet; diving rudder, 3 degrees dive; revolutions per minute, 154.33.

h. m. s.
2·54·10 Rose to surface. Maximum angle while rising, 6 degrees by stern.

THIRD RUN.

3·01·30 Order given to dive.
3·02·50 Reached 10 feet depth. Approximate conditions of steady motion: trim of vessel, 1 degree by head; depth by gauge, 11 feet; angle of diving rudder, 2 degrees dive; revolutions per minute, 177.33.
3·09·00 Order given to rise. Maximum angle of trim of vessel while coming to the surface, 5 degrees by stern.

FOURTH RUN.

3·13·30 Order given to dive.
3·14·10 Reached depth of 10 feet by gauge, the depth not, however, being maintained very steady, but varying between 5 and 15 feet by the gauge. While checking these oscillations the diving rudder was moved from an angle of 8 degrees dive to the maximum angle for rise permissible. Revolutions during this run, 176.28.
3·20·15 Order given to rise. Maximum angle of trim of vessel while rising, 9 degrees by stern.

FIFTH RUN.

3·28·35 Order given to dive. Depth during this run variable between 8 and 15 feet; revolutions per minute, 187.33.
3·35·30 Order given to rise.

SIXTH RUN.

3·42·30 Order given to dive.
3·42·40 Reached depth of 10 feet and then brought motor up to full speed. It is to be noted that while diving during both fifth and sixth runs the full speed of the motor was not used until the boat was steadied at a depth of about 10 feet below the surface, this being done on account of the diving rudder working hard. Revolutions per minute, 188.66.

h. m. s.
3·49·00 Rose to surface.
3·49·25 Order given to blow out midship tank.
3·50·25 Shut off midship tank.
3·50·26 Order given to blow out main tank.
3·54·30 Opening conning-tower top.
3·54·40 Shut off main tank.
3·56·00 Order given to blow out auxiliary tank.
4·00·00 Shut off auxiliary tank.

During these runs a small amount of water, between 8 and 10 buckets, collected in the forward part of the boat and just forward of the forward battery, apparently due to leak about valve fitted, pocketed, over the forward kingstons.

On May 18th the *Porpoise* ran her torpedo trial. On account of an error of judgment only one and a half miles were run, but the boat could just as well have run two miles. The breeze was light, and the sea smooth. The torpedo hit the target in the center.

SUBMARINE TORPEDO-BOAT *PORPOISE.*

NOTES TAKEN DURING THE TORPEDO TRIAL, MAY 17, 1903.

TIME.
h. m. s.
2·15·00 Got under way from deck. The voltage of the batteries on open circuit was as follows: After battery, 60 volts; forward battery, 59 volts; about 12 ampères were being used for illuminating purposes.
2·35·00 Preparing to put vessel in diving condition, sending below ventilators, etc.
2·40·05 Order given to fill midship tank.
2·40·50 Midship tank reported full.
2·42·20 Closed conning-tower top.
2·43·00 Order given to fill main tank.
2·46·00 Main tank reported full.
2·46·10 Order given to fill auxiliary tank.
2·50·10 Auxiliary tank shut off.
2·50·30 Order given to fill after trimming tank.

h. m. s.

2·52·50 Shut off after trimming tank.

2·52·50 Order given to fill auxiliary tank.

2·53·40 Order given to fill auxiliary tank slowly.

2·54·10 Shut off auxiliary tank. Shut off everything.

2·54·30 Started ahead on motor.

2·58·00 Order given to dive.

2·59·45 Depth of 10 feet reached, and started counter.

3·00·40 Depth gauge read 4 feet.

3·01·00 Depth gauge read 15 feet.

3·01·30 Depth gauge read 6 feet.

3·02·20 Depth gauge read 7 feet.

3·03·00 Depth gauge read 7 feet.

3·03·35 Depth gauge read 11 feet.

Approximate conditions of steady running at this time was as follows: Trim of vessel, 1 degree by stern; depth by gauge, 11 feet; angle of diving rudder, 4 degrees dive.

3·08·00 The vessel being at a depth of 10 feet, an order was given to rise. The vessel reached the surface 35 seconds after the order was given.

3·08·35 Reached surface.

3·09·00 Order given to dive. Angle of trim while diving, 7 degrees by head.

3·09·55 Reached depth of 9 feet.

3·10·10 Approximately steady motion at a depth of 12 feet.

3·11·06 Depth gauge read 6 feet.

3·11·30 Depth gauge read 10 feet.

3·11·45 Order given to "Get ready to fire."

3·12·30 Order given to rise to surface.

3·13·10 Vessel being on surface, order was given to dive.

3·13·30 Order given to rise to surface.

3·13·45 Order given to fire torpedo.

3·14·00 Order given to blow out midship tank.

3·15·00 Stopped motor.

3·19·00 The voltage of batteries on open circuit were as follows: Forward battery, 58 volts; after battery, 58½ volts.

The *Porpoise* was selected by the Board of Inspection and Survey to make the twelve-hours' run light under all cylinders of the gasoline engine. On the morning of May 21st she was secured with her nose against the piles of a dock, and held by a line from each quarter. The engine was started at 6 A. M. and ran until 4·30 P. M., when it was stopped, and the boat taken out for a run in the Bay. The engine was started again at 4·41 and ran until 6·11, when the twelve-hours' run was finished successfully. The mean revolutions alongside the dock were 203.67 and underway 248.9.

The engine worked well for the twelve hours. Alongside the dock all ventilators were shipped, the conning tower and for-

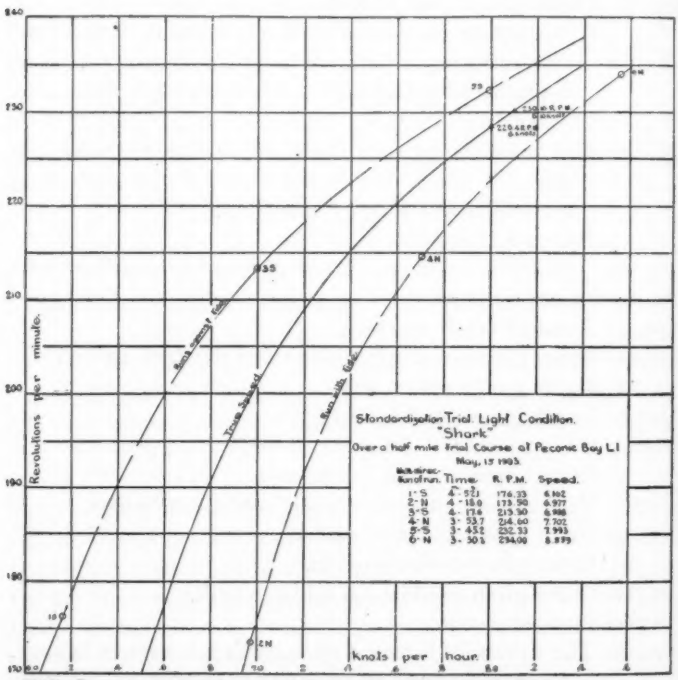


FIG. 2.—U. S. S. "SHARK," SUBMARINE BOAT. STANDARDIZATION TRIAL, LIGHT CONDITION, MAY 15, 1903.

ward hatch also being open. The air was good at all times, and no foul odors were noticed.

While underway, the forward hatch was closed. For a period of 37 minutes, the door of the conning tower was also closed, the only air entering coming through the ventilators. The temperature rose slightly, but the air remained good. An examination of the muffler the next day showed it to be in excellent condition.

SUBMARINE TORPEDO-BOAT *SHARK*.

On May 14th the *Shark* was successfully standardized awash, 196.6 revolutions per minute being necessary for seven knots, and a speed of 7.3355 being made on one double run. Immediately afterwards the *Shark* was standardized light, 228.4 revolutions per minute being necessary for eight knots, and a mean

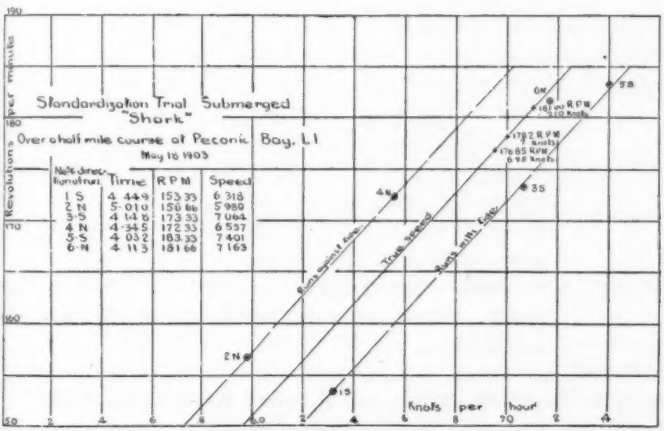


FIG. 3.—U. S. S. "SHARK," SUBMARINE BOAT. STANDARDIZATION TRIAL, SUBMERGED, MAY 18, 1903.

speed of 8.276 knots being made on one double run. The revolution speed curve for this trial is shown in Fig. 2. The next day, May 15th, the *Shark* finished successfully her ten-mile run awash. The run lasted one hour and a half, the mean revolutions being 209.51, and the mean speed 7.261 knots per hour.

On May 18th the *Shark* was standardized submerged. For a speed of 7 knots 178.3 revolutions per minute were found necessary. The water was smooth and breeze light. The revolution speed curve of this run is indicated in Fig. 3.

SUBMARINE TORPEDO-BOAT *SHARK*.

NOTES TAKEN DURING THE STANDARDIZATION TRIAL, SUBMERGED, MAY 17, 1903.

Before beginning the trial the readings of voltage of the storage batteries on open circuit were taken and found to be as follows: Forward battery, 60 volts; after battery, 60 volts.

TIME. h. m. s.	
9·48·00	Order given to fill main tank.
9·49·55	Shut off main tank, it being reported full.
9·50·00	Order given to fill forward trimming tank.
9·55·00	Shut off forward trimming tank.
9·57·00	About two quarts of gasoline were found in the forward bilge of the boat. This leaked out of the gland of the glass gauge tube, which had not been properly set up; opened the conning tower top and, after taking up this gasoline in a bucket with a sponge, the gasoline was thrown overboard and the gland on gauge glass carefully set up.
10·00·45	Shut down conning-tower top and went ahead on the motor. Strong odor of gasoline noticeable throughout the boat.
10·02 to 10·05·00	Repeated starting and stopping motor to test trim.
10·05·10	Order given to fill auxiliary tank.
10·11·00	Auxiliary tank reported full.
10·11·10	Order given to fill forward trimming tank.
10·11·40	Forward trimming tank reported full.
10·11·45	Order given to fill compensating tank under torpedo tube.
10·16·10	Compensating tank under tube reported full.
10·16·15	Order to fill after trimming tank.

h. m. s.
10·20·20 Shut off after trimming tank.
10·20·35 Order given to fill gasoline tank (with water, to gain additional trim by head).
10·21·35 Order given to shut off everything.
10·22·00 Started ahead on electric motor.

FIRST RUN.

10·24·00 Order given to dive.
10·25·00 Reached 10 feet depth of water by gauge. Conditions of steady motion: trim of vessel, 2½ degrees by head; depth by gauge, 10 feet; angle of diving rudder, 3½ degrees dive; revolutions per minute, 153.33. Another observation for conditions of steady motion gave—trim of vessel, 4 degrees by head; depth by gauge, 10 feet; diving rudder, 2 degrees dive.
10·31·20 Rose to surface. Angle of diving rudder while rising to surface was set at 1 degree rise until the depth gauge showed about 5 feet, when the angle of rudder was increased to 8 degrees rise.
While running on surface, turning, the trim varied between 2 and 5 degrees by stern; depth gauge reading 2½ feet; and the angle of diving rudder varying between 2 and 4 degrees rise.

SECOND RUN.

10·37·30 Order given to dive. While going down, the diving rudder, after the first impulse in starting down had been given, was kept at an angle of about 8 degrees rise, to prevent too great an angle of trim being reached in diving.
10·39·00 Reached depth of 10 feet by gauge. Approximate conditions of steady motion: trim of vessel, 2½ to 3 degrees by head; depth by gauge, 10 to 10½ feet; diving rudder, 3 to 4 degrees dive; revolutions per minute, 156.66.

h. m. s.

10·45·00 Rose to surface. While rising, the diving rudder was kept at 2 degrees rise until near the surface, when it was run up to 8 degrees rise; the angle of trim of boat while running on the surface with this angle of diving rudder being 5 degrees by stern. While running on the surface the following conditions of steady motion were noted: trim of vessel, 4 degrees by stern; depth gauge read 2½ feet; diving rudder, 3 degrees rise.

THIRD RUN.

10·51·40 Order given to dive.

10·53·00 Reached depth of 10 feet. Conditions of steady motion: trim of vessel, 2½ to 3 degrees by head; depth, 9½ to 10½ feet; angle of diving rudder, 2 to 3 degrees dive; revolutions per minute, 173.33.

10·59·00 Rose to surface. While coming to surface the diving rudder was kept at 2½ degrees rise until the surface was reached, when it was brought to 9 degrees rise, which gave a trim of 5 degrees by the stern. While running on the surface the following conditions were noted: Approximate conditions of steady motion: trim of vessel, 6 degrees by stern; depth by gauge, 3 feet; angle of diving rudder, 1 degree rise.

FOURTH RUN.

11·04·00 Order given to dive. Vessel went down at a maximum angle of 9 degrees by the head. The rudder was brought to an angle of 14 degrees rise. Approximate conditions of steady motion: trim of vessel, 2 to 3 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 2 to 3 degrees dive; revolutions per minute, 172.33.

11·08·30 Considerable amount of smoke was noticed in vicinity of motor, apparently due to heating of "A" frame bearing of main shaft immediately above armature.

h. m. s.
11·10·00 Rose to surface. Maximum angle of trim of vessel while rising, 8 degrees by stern.
11·15·00 Felt motor commutator by hand; it seemed somewhat heated, but no opportunity for taking exact temperatures was afforded.

FIFTH RUN.

11·20·00 Order given to dive.
11·21·00 Reached depth of 12 feet. Approximate conditions of steady motion: trim of vessel, 2 degrees by head; depth, 10 feet by gauge; diving rudder, 2 degrees dive; revolutions per minute, 183.33.
11·26·00 Rose to surface. Time taken to reach surface from depth of 11 feet, 30 seconds.
While running on surface, approximate conditions of steady motion: trim of vessel, 6 degrees by stern; depth by gauge, 3 feet; angle of diving rudder, 2 degrees rise. Considerable amount of thin smoke noted throughout the boat during this run, and throughout the remainder of the trial.

SIXTH RUN.

11·32·00 Order given to dive.
11·33·00 Reached depth of 10 feet. Maximum angle of trim of vessel while going down, 9 degrees by head. Approximate conditions of steady motion: trim of vessel, 3 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 2 degrees dive; revolutions per minute, 181.66.
11·39·00 Rose to surface. Time taken to rise from 10 feet depth to surface, 15 seconds.
11·40·00 Order given to blow out midship tank.
Opened conning-tower top.
11·40·30 Shut off midship tank.
11·40·45 Order given to blow out main tank.
11·44·30 Started putting up after ventilator.

h. m. s.
11·46·00 Started after ventilating fan motor.
11·46·45 Shut off main tank.
11·47·00 Order given to blow out auxiliary tank.
11·48·00 Started ahead on propelling motor.
11·49·45 Shut off auxiliary tank.
11·49·50 Order given to blow out forward trimming tank.
11·53·30 Shut off trimming tank.
The smoke noticed during this trial was apparently due to oil from a piece of canvas that had been placed about the main shaft, over the propelling motor, to prevent the oil reaching the commutator; the oil, dripping on this canvas, had passed through it, a certain amount falling on the commutator.
11·55·00 Considerable amount of smoke remaining in the after part of the boat.
11·56·00 Reversed the motor operating the small Root blower ventilator of after part of boat, so as to blow air in the boat.

The *Shark* had her torpedo trial on May 19th, the three-hour endurance run to follow immediately after. On account of the heating of the after main bearing the run was discontinued after firing the torpedo, which hit the target. The speed fell below the seven knots required—6.95 knots.

SUBMARINE TORPEDO-BOAT *SHARK.*

NOTES TAKEN DURING THE TORPEDO TRIAL, MAY 19, 1903.

TIME.
h. m. s.
2·19·00 Left the dock, being towed out by steamer *Kelpie.*
2·35·00 Began to prepare for diving, sending ventilators below, etc.
2·44·00 Put the torpedo in tube, the conning-tower hatch being then open. Temperature of thermometer hung along side of depth gauge, 80 degrees Fahrenheit; barometer, 30.34 inches. Took voltage of battery on open circuit with the following results: forward battery, 61 volts; after battery, 60⅜ volts.

h. m. s.
2·50·00 Order given to fill main tank.
2·51·00 Main tank reported full.
2·51·05 Order given to fill forward trimming tank.
2·56·45 Shut off forward trimming tank.
2·57·00 Order given to fill auxiliary tank.
2·59·25 Order given to fill forward trimming tank.
3·01·25 Shut off forward trimming tank.
3·02·00 Shut off auxiliary tank.
3·02·20 Order given to fill compensating tank under torpedo tube.
3·02·40 Compensating tank reported full.
3·02·45 Order given to fill oil tank (with water, for additional weight forward).
3·04·20 Shut off oil tank.
3·04·45 Order given to fill auxiliary tank slowly.
3·05·20 Shut off auxiliary tank and everything.
3·05·30 Order given to go ahead on motor.
3·10.00 Order given to dive.
3·11·30 Depth gauge registered 12 feet; thermometer, 82 degrees Fahrenheit; barometer, 30.40 inches. Approximate conditions of steady motion: trim of vessel, 3 degrees by head; depth gauge read 13 feet; angle of diving rudder, 4 degrees dive. There were 12 persons in the boat—7 of the crew and 5 observers.
3·24·00 Rose to the surface.
3·24·25 Order given to dive.
3·25·00 Depth gauge read 8 feet.
3·26·00 Order given to "Get ready to fire."
3·27·25 Rose to surface.
3·27·45 Order given to dive.
3·28·20 Rose to the surface.
3·28·40 Order given to "Fire torpedo."
3·29·00 Order given to blow out midship tank.
3·29·40 Order given to shut off midship tank.
3·30·20 Order given to blow out main tank.
At this time the thermometer read 78 degrees Fahrenheit.

It had been the intention of the contractors to make this trial part of a three-hour endurance trial, but on account of the "A" frame bearing of main shaft, located over the electric motor, heating badly during this trial, the endurance trial was discontinued.

The three-hours' endurance run was made by the *Shark* on May 20th. Besides the crew of seven men, there were five observers in the boat. The three-hours' run was successfully finished under some disadvantages. The "A" frame bearing of the main shaft heated slowly, and at the end of an hour and a half was smoking.

There was an overload on the motor, which the contractor said was due to the thrust being too far forward, allowing the screw to rub against the hull. The gear wheel, through which the motor turns the shaft, was rubbing against the "A" frame bearing of the shaft, heating it badly. This last must have been the cause of some of the overload. The "A" frame bearing became too hot to touch, and the motor and commutator were nearly as hot. At the end of the run the boat was filled with smoke from the hot bearing. The air was not foul, and the only discomfort experienced was a slight smarting of the eyes.

The mean speed was 7·1 knots, the average revolutions, 181.01.

NOTES TAKEN DURING THE THREE-HOUR SUBMERGED ENDURANCE TRIAL, MAY 20, 1903.

TIME. h. m. s.	
12·48·00	The *Shark* left the dock in tow of the steamer *Kelpie.*
1·05·00	Cast off tow line, and started, sending ventilators, etc., below, preparing the vessel for diving.
1·06·00	Order given to fill torpedo tube.
1·10·45	Order given to fill midship tank.
1·11·10	Shut off midship tank.
1·11·25	Order given to fill main tank.
	The voltage of the storage batteries was taken on open circuit and found to be as follows: Forward battery, 61 volts; after battery, 61 volts.
	The thermometer and barometer had been hung alongside the depth gauge, and at this time read: barometer, 30.30 inches; thermometer, 84 degrees Fahrenheit.

h. m. s.

1·13·10 Order given to fill auxiliary tank.

1·13·30 Shut off main tank.

1·17·10 Shut off auxiliary tank.

1·17·20 Order given to fill oil tank.

1·17·50 Shut off oil tank.

The boat was then ready to dive, but was waiting to have the steamer *Kelpie* get off the course. Sixteen ampères were being used for lights during this run.

1·22·15 Started ahead on motor, but stopped almost immediately on account of the torpedo boat *Craven* getting on the course. While at rest the trim of vessel was 1 degree by the head.

1·27·00 Started ahead on motor.

1·27·50 Order given to dive.

1·29·00 Depth gauge registered 10 feet.

1·29·30 Began endurance trial. The conditions of steady motion at this time were: trim of vessel, 3½ degrees by head; depth gauge read 10 feet; angle of diving rudder, 3 degrees dive.

1·35·00 Depth gauge read 10 feet.

1·39·30 Rose to surface.

1·40·50 Order given to dive.

1·46·00 Depth gauge read 15 feet.

1·46·30 Depth gauge read 7 feet.

1·47·15 Rose to surface.

1·48·10 Dove. At this time barometer read 30.40 inches; thermometer, 80 degrees Fahrenheit.

1·50·00 Depth gauge read 10 feet.

1·55·00 Depth gauge read 10 feet.

1·57·10 Rose to surface. (Time to come to surface from 10 feet depth, 15 seconds.)

2·00·00 At this time barometer read 30.48 inches; thermometer, 78 degrees Fahrenheit.

2·02·30 Order given to dive. At this time, while boat was running on the surface just before diving, 520 ampères were being used, which ran up to 570 while diving and fell to 505 while running submerged at a depth of 10 feet.

h. m. s.

During this trial there were 12 men in the boat—7 belonging to the crew and 5 observers.

2·07·35 Rose to surface.

2·07·45 Dived to a depth of 7 feet and then rose to surface immediately, diving immediately thereafter. The motion of the boat at this time was very irregular.

2·08·50 Order given to dive.

2·09·45 Reached depth of 9 feet. At this time the motion of the boat had become fairly uniform. An endeavor was made to determine the effect upon the angle of the diving rudder of moving the men about within the boat. A man weighing 140 pounds was moved from a point just forward of the gasoline engine to the extreme forward end of the vessel, the resulting change in the angle of the diving rudder being less than 2 degrees. The change in the position of the diving rudder due to the normal conditions of balancing the excess buoyancy by the tendency to dive due to the diving rudder, was 5 to 6 degrees, the rudder being moved from approximately an angle of about 2 degrees rise to 4 degrees dive. The movement of the men within the boat was, therefore, negligible, and, as a matter of fact, men were moved about in the boat at will, the only necessary precaution being to inform the man at the diving rudder that change in position was to be made, and to move with moderate speed from point to point.

2·20·00 Conditions of steady motion noted: trim of vessel, 3 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 2 degrees dive.

2·25·00 Rose to surface.

2·25·45 Order given to dive.

2·29·30 Depth gauge read 11 feet; barometer, 30.66 inches; thermometer, 77 degrees Fahrenheit.

2·32·30 Rose to surface. While running on the surface, 550 ampères were used.

2·37·00 Order given to dive. The vessel dived very irregularly, "porpoising" between 5 and 15 feet depth by gauge.

h. m. s.

2'39'00 Boat steady at 10-foot depth. Barometer, 30.74 inches.

2'48'00 Rose to surface.

2'49'30 Order given to dive.

2'50'15 Depth gauge registered 10 feet. The "A" frame bearing of main shaft over motor began heating and gave off a light, thin smoke, which spread throughout the boat and was somewhat irritating to the eyes and nose.

2'55.10 Rose to surface.

2'59'55 Order given to dive.

3'04'00 Barometer, 30.52 inches; thermometer, 78 degrees Fahrenheit; depth gauge read 12 feet. Conditions of steady motion noted at this time: trim of vessel, 3 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 2½ degrees dive.

3'14'30 Depth gauge read 10 feet; barometer read 30.58 inches; thermometer, 78½ degrees Fahrenheit.

3'25'30 Order given to dive.

3'29'30 Depth gauge read 10 feet; barometer, 30.80 inches; thermometer, 79.5 degrees Fahrenheit. Boat not running steadily; moderate oscillation of the diving rudder between angles of 6 degrees rise and 8 degrees dive.

3'32'00 Rose to surface.

3'34'00 Order given to dive. While diving, maximum angle of trim of vessel was 12 degrees by head.

3'35'10 Vessel steadied at 13 feet depth by gauge. Conditions of steady motion: trim of vessel, 4 degrees by head; depth by gauge, 13 feet; angle of diving rudder, 3 degrees dive.

3'40'30 Rose to surface.

3'44'00 Order given to dive. Conditions of steady motion noted: trim of vessel, 4 degrees by head; depth by gauge, 12 feet; angle of diving rudder, 3 degrees dive.

3'50'00 Vessel oscillating moderately, trim varying from 4 to 9 degrees by head; depth by gauge from 10 to 12 feet; angle of diving rudder, from 2 to 4 degrees dive; barometer, 30.72 inches; thermometer, 80 degrees Fahrenheit.

h. m. s.

The "A" frame bearing on main shaft, over the motor, was heating considerably, giving off smoke and considerably heating the after part of the vessel. A thermometer hung against the skin plating abreast this motor at this time read 97½ degress Fahrenheit.

3·52·00 Conditions of steady motion noted: trim of vessel, 4 degrees by head; depth by gauge, 10 feet; angle of diving rudder, 2 degrees dive.

3·57·00 Rose to surface. Barometer, 30.76 inches; thermometer, 80 degrees Fahrenheit. Smoke in boat, due to burning of oil in after bearing, somewhat increasing; smarting sensation noted in the eyes from the effect of smoke. At this time, while running on the surface, the motor used 610 ampères.

4·03·20 Order given to dive. While diving at this time the current rose to 750 ampères. Maximum angle of trim of vessel while going down, 10 degrees by head.

On looking through the periscope the shore line was lost in the haze and very indistinct, but the water in the near foreground was very clear, the use of a periscope at this time on objects in the distance being very unsatisfactory on account of the natural atmospheric conditions then existing, there being considerable haze and smoke from forest fires hanging over the water.

4·10·00 Thermometer hanging against outside plating, opposite the motor, read 96 degrees Fahrenheit. Motor frame and field coils were very hot when felt, but no appliance available for taking their temperatures.

4·13·30 Depth by gauge, 10 feet; barometer, 30.74 inches; thermometer, 80 degrees Fahrenheit.

4·14·00 Rose to surface.

4·16·30 Order given to dive. Boat went down to a depth of 20 feet, then came immediately to the surface, "porpoising" violently, so as to render the following measurements somewhat uncertain.

4·17·00 Rose to surface.

h. m. s.
4·18·30 Going down.
4·18·45 Depth gauge read 6 feet.
4·18·50 Depth gauge read 12 feet.
4·19·40 Depth gauge read 4 feet. Man at periscope reported to Captain that the *Craven* was directly ahead.
4·20·00 Running on surface.
4·23·00 Barometer, 30.74 inches; thermometer, 80 degrees Fahrenheit.
4·24·00 Vessel still running on surface.
4·27·00 Order given to dive.
4·29·30 Depth gauge read 7 feet; barometer, 30.71 inches; thermometer, 89 degrees Fahrenheit. Three-hours' endurance trial completed.
4·30·00 Blew out midship tank. Thermometer opposite motor read 100 degrees Fahrenheit. Inside of boat was full of smoke from burning oil on the "A" frame bearing over the motor.

The motor being stopped, the voltage of the storage batteries was taken on open circuit and found to be as follows: forward battery, 57 volts; after battery, 57 volts.

Observers on the tug stated that the *Shark* was trimmed for this trial with an excess of buoyancy, and that she shot up like a porpoise when she arose and seemed to go under with difficulty. The current rose fifty or more ampères on every dive, and at the end of the trial 700 ampères were necessary on diving. In spite of the severe demands made upon it, the battery was not run down at the end of the trial. If everything in the boat is in good condition, the battery should be able to drive her for nearly four hours at a speed of seven knots.

The weather during the trial was hazy, and the water smooth. Owing to the difficulty of seeing, the boat rose to the surface a great deal, so only about two hours were spent beneath the surface.

On May 21, 1903, the *Shark* made her ten-mile run in light condition. Mean revolutions, 230.16; mean speed, 8.10 knots; duration of trial, 1 hour 20 minutes.

CONCLUSIONS.

In the foregoing pages little attention has been paid to surface runs, as the water was never rough and the boats were only doing what any gasoline launch could do. But little new information was obtained in these submarine runs, and no opinion could be formed as to what the boats would do in rough water. Although there were five extra men in the boat, the air was not foul at the end of three hours.

An experiment with a composition called "Anti-Sweat," by applying it on both sides of the glasses in the conning tower, showed that it could neither keep off the spray nor the moisture from the helmsman's breath.

The machinery in the boats is excessively crowded and poorly arranged. On the *Porpoise* the coupling running the bilge pump slipped and was most difficult to get at; an expert contortionist would be required to overhaul or repair it with any facility. The thrust bearing can hardly be reached on account of the levers being in the way. The end of a main bearing projects over the commutator, dripping oil on it, causing smoke and sparking. There is not enough clearance between the different parts, and the least maladjustment of the clutches or wear in the thrust brings the parts together, resulting in hot bearings, smoke and overload on dynamo. The machinery can be arranged in much better shape.

A PROBLEM CONNECTED WITH THE SCREW PROPELLER.

By W. F. Durand, Member.

The usual problem involved in the design of a screw propeller calls for the determination of the dimensions of the propeller, having given the horsepower to be developed by the engine, and the speed to be attained by the boat. In brief, the power and speed are given, and the propeller is required. In connection with various questions relating to screw propulsion, however, there may arise the inverse problem as follows: given the ship, the propeller, the engine and the steam pressure to be maintained by the boilers, what will happen? More exactly, the problem may be stated as: given the ship and propeller and the turning moment on the shaft, what will be the revolutions and speed?

In order to properly appreciate the character of this problem, it may be well at this point to call attention to the two following propositions:

(1) The work developed by the engine is measured by the product of the turning moment, by the angular velocity, or by 2π times the revolutions.

(2) The schedule of steam pressures through the engine fixes the turning moment, and, in consequence, so long as the steam pressure remains unchanged the turning moment will remain the same.

It will thus be noted that fixed steam pressure and fixed cut-offs mean constant indicator cards and a constant turning moment, while the power developed will be measured by the product of this moment by 2π times the particular revolutions which the combination of conditions will permit. It will be noted in particular that in the problem as stated the power is not known and cannot be considered as a known quantity, since it

depends on the revolutions, one of the unknown conditions to be determined.

A problem of this character may arise in connection with the combination of a given propeller with a given ship, engine and boilers, the latter of which are supposed to be able to maintain a given pressure. Again: with given ship, engines and boilers capable of maintaining a given pressure, a propeller is designed and fitted and a given result is produced. An estimate is then desired of the consequences of changing the propeller slightly, with the assumption of fixed steam pressure and turning moment. It may be that there is evidence that the pitch is too high, the revolutions too low, and that in consequence the full power is not developed. It may then be proposed to keep the diameter the same, but to decrease the pitch and thus allow the revolutions to increase, which, with constant turning moment, will give increased power. The resulting revolutions and speed are then desired. Again; suppose instead of changing the pitch everything be kept constant but the area, which is reduced so as to allow the revolutions to increase. What in such case will be the result?

Such problems may arise in many forms, but all involve similar fundamental conditions, and in none can the *power* be properly considered as one of the known quantities.

A very brief examination will serve to show that none of the formulæ for propeller design commonly employed and involving the power will serve to deal with the problem here proposed. It will therefore be necessary to take such equations as may be available for the purpose, and seek for such transformation as may be suited to the solution of the problem in the form stated.

We shall use the following notation:

W = indicated power.
Wp = power delivered to the propeller shaft.
f = mechanical efficiency of engines $= Wp \div W$.
U = net work developed by propeller.
e = efficiency of propeller.
A = area of L.P. piston of engine.

p_m = mean effective pressure reduced to L.P. piston.
Q = turning moment on propeller shaft.
T = thrust.
R = resistance.
d = diameter of propeller.
p = pitch of propeller.
N = revolutions.
u = speed in knots.
s = slip.
k = factor in propeller equation denoting influence of slip.
l = factor denoting influence of pitch ratio.
im = combined factor denoting influence of shape and area of blades.
h = $k \div (1 - s)$.

Then in general we have:

$$Q = f_1(d\ p\ N\ u\ im). \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$T = f_2(d\ p\ N\ u\ im). \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$R = f_3(\text{ship, speed}) = T. \quad . \quad . \quad . \quad . \quad . \quad (3)$$

These equations, general in form and without attempt to express relations in detail, imply simply that turning moment, or torque, and thrust are both determined conjointly by the dimensions and characteristics of the propeller, by the revolutions and by the speed; also that resistance depends on the ship and on the speed, and that the thrust must be its equal. If, now, we can suppose the forms of equations (1) and (2) known, then we can suppose N to be eliminated between them, leaving us an equation involving Q, T, d, p, u and im. This we may express in the form

$$T = F(Q\ d\ p\ u\ im). \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Now in the problem assumed, the propeller is fixed in dimension and feature, and the ship as well. Hence the only things remaining unknown and variable are N and u, revolutions and speed.

We may therefore rewrite the above equations as follows:

$$Q = f_1(Nu), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$T = f_2(Nu), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$R = f_3(u). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and by elimination between (5) and (6),

$$T = F(Q\,u). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

We have now only to remember that $T = R$, and from (7) and (8) determine the resulting value of u.

This will serve, as we shall see later, to determine N and any other features which may be desired. This shows the logical development of the solution. Actually there are many special points of detail to which we may now give attention.

In the first place, none of the ordinary equations or formulæ relating to propeller design or to its performance are in form as called for by (1), (2), (4), (5) or (6). Such equations may, however, be developed by suitable procedure, the steps of which we will next examine.

The fundamental equation for the propeller may be expressed as follows:

$$U = d^2\,(p\,N)^3\,k\,l\,im.^* \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

This equation is equivalent to the assumption that the work varies as the square of the linear dimensions, as the cube of the speed, and, furthermore, depends on the *slip*, *pitch ratio*, *shape* and *area* of blade.

This general foundation is quite uncontested, and, whatever may be the details of propeller theory, it is unquestioned that the characteristics of the propeller affect the performance in the general manner indicated by this equation, and that the present great need is the establishment of facts which would serve to give more reliable numerical values to the factors k, l, im. The point of importance here is that, quite independent of the views of any particular individual regarding the details of propeller

* For the derivation of this equation see Author's text book on "Resistance and Propulsion of Ships," Chapter IV.

theory or the best form of such equations and formulæ, there is a general agreement regarding the character of the relations between the useful work and the controlling elements of the propeller and of its use, and this general agreement is embodied in one of its forms in the equation (9) above.

Now if s denote the slip we have:

$$u = p\,N(1-s) \div 101.3$$

and

$$T = U \div 101.3\,u = U \div p\,N(1-s). \quad . \quad . \quad . \quad (10)$$

It will be remembered that the factor k is a function of s alone. Then if we put $k \div (1-s) = h$, we shall have from (9) and (10)

$$T = d^2\,p^2\,N^2\,h\,l\,im. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Also let Wp denote the work delivered to the propeller, e the propeller efficiency, f that of the engine, and W the total I.H.P., then

$$Wp = f\,W = U \div e,$$

also

$$Wp = 2\pi\,Q\,N.$$

Hence

$$Q = \frac{Wp}{2\pi\,N} = \frac{f\,W}{2\pi\,N} = \frac{U}{2\pi\,N\,e} = \frac{d^2\,p^3\,N^2\,k\,l\,im}{2\pi\,e}. \quad . \quad (12)$$

Also $33{,}000\,W = 2p_m\,L\,A\,N$, foot pounds per minute.

Hence

$$Q = \frac{33{,}000\,f\,W}{2\pi\,N} = \frac{f\,p_m\,L\,A}{\pi} \text{ pound-feet}, \quad . \quad . \quad (13)$$

or

$$Q = \frac{f\,p_m\,L\,A}{2{,}240\,\pi} \text{ ton-feet}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The value of the efficiency e depends primarily on the pitch ratio $p \div d$ and the slip ratio s. The former is fixed by the propeller, the latter by the relations between revolutions and speed.

With the given propeller, therefore, the former is fixed, while the latter remains unknown. Hence e is an unknown quantity, and in a rigorous solution must be treated as such. It is fortunately the fact, however, that the efficiency does not vary

rapidly for considerable change in the value of the slip, and hence, while the latter may be unknown and with it the value of e, nevertheless the problem may be very satisfactorily treated by taking first an assumed value for e and carrying the solution through on this assumption. The resulting conditions may then indicate another value for e, by means of which, if desired, a second approximation may be made in the results obtained. In many cases, however, no second approximation will be required, and the value of e may safely be assumed; but the value thus taken must be applicable to the general conditions of the problem, and in all usual cases will be found between .66 and .70.

Turning again to equations (11) and (12), it will be noted that in the expressions on the right-hand side the only quantities remaining variable and unknown in the case of a given propeller are N, k and h. Let us write these equations as follows:

$$\frac{T}{d^2 p^2 l\, im} = N^2 h. \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$\frac{2\pi Q e}{d^2 p^3 l\, im} = N^2 k. \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

The expressions on the left may be called, respectively, the *thrust constant* and *torque constant.* Let these be denoted by T_1 and Q_1, respectively. Then we have

$$T_1 = N^2 h, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$Q_1 = N^2 k; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

whence

$$\frac{Q_1}{T_1} = \frac{k}{h} = (1 - s), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

and

$$s = 1 - \frac{Q_1}{T_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

Again

$$u = p\, N(1 - s) \div 101.3, \quad . \quad . \quad . \quad . \quad . \quad (21)$$

and

$$\frac{101.3\, u}{p} = N(1 - s). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

The ratio $101.3u \div p$ may be called the speed constant, and represents simply the ratio of speed to pitch. This quantity we shall use as the variable representing speed, and shall denote it by x_1. Then $x_1 = N(1-s)$.

Now investigation shows that the values of h as a function of s may be very closely expressed by a simple linear equation of the form

$$h = r\,s + q\,; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

hence from (20)

$$h = r\,s + q = r\left(1 - \frac{Q_1}{T_1}\right) + q. \quad . \quad . \quad . \quad . \quad (24)$$

We are now ready to build up an expression for $N(1-s)$ which shall contain only $Q_1\ T_1\ r$ and q.

From (17) we have

$$N = \sqrt{\frac{T_1}{h}},$$

and from (19)

$$(1-s) = \frac{Q_1}{T_1}.$$

Hence

$$N(1-s) = \frac{Q_1}{T_1}\sqrt{\frac{T_1}{h}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

or

$$N(1-s) = \frac{Q_1}{T_1}\sqrt{\frac{T_1}{r\left(1-\frac{Q_1}{T_1}\right)+q}}. \quad . \quad . \quad . \quad . \quad (26)$$

Putting x for $N(1-s)$ and reducing, we find:

$$T_1 = \frac{Q_1^{\,2}}{(r+q)\,x_1^{\,2}} + \frac{r}{r+q}\,Q_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

This equation, it is noted, expresses T as a function of $Q_1\ x_1$ and the numerical coefficients r and q; or, in other words, thrust is thus expressed as a function of torque and speed.

The above transformation from equation (18) may be summarized under the following steps:

(1) The selection of the quantity $N(1-s)$, proportional to speed, and the adoption of this as a new variable.

(2) The expression of the value of this variable in terms of Q_1,

T_1 and numerical coefficients, and the consequent elimination of N, k and h from the equations (17) and (18).

We may next turn to the question of the numerical values of r and q.

By reference to such experimental results as are available, and using feet and pounds as units, it may be shown that the value of h may be empirically expressed by

$$h = .000316\,s + .000006, \quad . \quad . \quad . \quad . \quad . \quad (28)$$

whence $r = .000316$ and $q = .000006$.

Substituting these values, we then have:

$$T_1 = \frac{Q_1^2}{.000322\,x_1^2} + \frac{.000316}{.000322}\,Q_1. \quad . \quad . \quad . \quad . \quad (29)$$

or the equivalent:

$$T_1 = 3{,}100\,\frac{Q_1^2}{x_1^2} + .98\,Q_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

The equation in this form involves T_1, x_1 and Q_1, and is thus explicitly free of the dimensions of the propeller. The relation between the various quantities involved are thus expressed in the terms of what may be considered as thrust speed and torque constants or factors as defined in (15), (16) and (22). For certain purposes, however, it may be of advantage to have the various dimensions and other quantities explicitly involved in the equation. To this end we may substitute in (30) for T_1 and Q_1, their values from (15) and (16) and take for the abscissa $x = pN(1 - s)$, or the speed in feet per minute. Then:

$$x = px_1 \text{ or } x_1 = x \div p.$$

Making these substitutions in (30) and reducing, we find

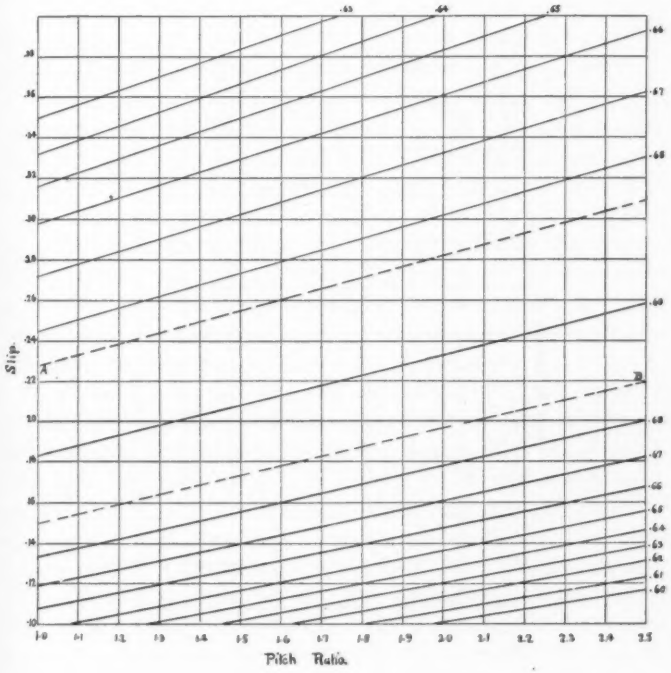
$$T = \frac{122{,}400\,e^2}{d^2\,p^2\,l\,im}\,\frac{Q^2}{x^2} + \frac{6.16\,e}{p}\,Q. \quad . \quad . \quad . \quad . \quad (31)$$

In practical use it will be found more convenient to use the *ton* as the unit of force and *10 feet* as a unit for d, p and $101.3u$. Making the corresponding changes in the numerical factors we have

$$T = \frac{274\,e^2}{d^2\,p^2\,l\,im}\,\frac{Q^2}{x^2} + \frac{.616\,e}{p}\,Q. \quad . \quad . \quad . \quad (32)$$

For the values of the efficiency we may be guided by the diagram of Fig. 1, in which, based on Froude's experimental data, the efficiency is expressed as a function of pitch ratio and slip.

Fig 1.



For the value of l, we have from the same data the equation

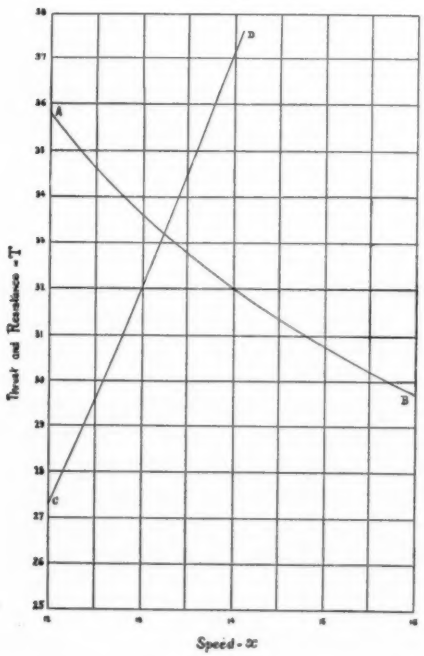
$$l = \frac{d}{p} - .17.$$

For the values of i and m we have but a small amount of data at present available. For areas of blade of about .36 disk area and of oval shape, $im = 1$. For some suggestions regarding

values for other conditions reference may be made to the author's text book on "Resistance and Propulsion of Ships," Chapter IV.

Suppose, now, that we have a given ship with a given turning moment on the shaft, and let the curve AB of Fig. 2 represent the corresponding relation between T and x in (32).

Fig 2.



Suppose, next, that we plot a curve CD between T and x for the ship. This would be in effect a curve of resistance plotted on speed, and would therefore presumably result most accurately from a series of model experiments. If such are not at hand, however, the values for the curve must be derived by such formulæ as may be considered suitable. It will be noted that the entire curve is not needed, but only that part which lies in the vicinity

of the curve AB, thus locating the intersection of the two. The values of the speed and thrust T thus become known. Then from (15) and (16) N^2h and N^2k may be found, and thence by (19) we find $(1-s)$. From the known values of p and of $pN(1-s)$, or speed, we then find the remaining unknown N, or revolutions.

The result of these operations is thus seen to determine, on the assumptions made, the revolutions and speed at which the various conditions will be fulfilled, and the problem is thus solved.

We may note here the method of allowing for the effect of wake and augmentation of resistance. In the various equations above, T and Q are the actual quantities developed, while x is the speed of the propeller through the water in which it operates. In all diagrams representing equations (30) or (32), therefore, it must be remembered that the speed involved is that of the propeller through the wake, and that in the expression $x=pN(1-s)$ the slip s is the so-called *true*, and not the *apparent*, slip. In plotting the values of the ship resistance curve, therefore, the following routine must be followed:

If w denotes the wake speed as a fraction of the speed of advance through the wake, then the ratio of the speed through the water to the speed through the wake will be $(1+w)$: 1. We then take a series of speeds through the wake and multiply them by $(1+w)$ to find the corresponding speeds through the water. We may then compute the resistances at these latter speeds. Computing by the usual formulæ or methods, this will give the true or tow-line resistance. These values must then be multiplied by the ratio of the augmented to the true resistance, in order to find at these speeds the actual thrust-block load. In default of special information, this ratio may be taken as approximately equal to $(1+w)$ the wake factor. The computed values of the resistance, multiplied by this factor, are then to be plotted on the speeds through the wake from which the original start was made. The speed given by the intersection will be the speed through the wake at which the engine, the ship and the propeller are in balance, and this speed multiplied by $(1+w)$ will give the speed through the water, as desired.

For illustration in detail let us take the following example:

Length of ship, feet,	400
Beam, feet,	50
Draught, feet,	21
Displacement, tons,	8,000
Diameter of propeller, feet,	17
Pitch, feet,	22
Diameter of L.P. cylinder, inches,	87
Stroke, inches,	54

Suppose boilers capable of maintaining a reduced mean effective pressure at the L.P. of 35 pounds per square inch. Suppose also that the conditions which are expected to result would justify an assumption of a propeller efficiency of .68. We will also take the mechanical efficiency of the engine as .87 and the wake factor as .14 for a single-screw ship as assumed.

First from (14) we have

$$Q = \frac{.87 \times 35 \times 4.5 \times 5.945}{2,240\,\pi} = 115.76.$$

Next substituting in (32) we have

$$T = \frac{201.300}{x^2} + 22.04.$$

For values of x we will take 12, 13, 14, 15 and 16 knots, reduce these to feet per minute and divide by 10, in accordance with the units for which (32) is arranged. This gives us for x the numbers 121.5, 131.7, 141.9, 152 and 162.1. Substituting and reducing we find for T the values 35.68, 33.65, 32.04, 30.75 and 29.70. These being plotted give the curve AB of Fig. 2. We next take the same speeds through the wake, and find for the equivalent speeds through the water the values 13.68, 14.82, 15.96, 17.10 and 18.24. We then compute first the tow-line and then the augmented resistances at these speeds. With the formulae actually employed the following values were found: 27.30, 32.03, 37.16, 42.67 and 48.55. These values being plotted on the speeds through the wake give the curve CD. At the intersection we

have speed = 13.25 and $T = 33.2$. We then readily derive the following results:

u through wake	= 13.25.
$p\,N(1-s) = 101.3 \times 13.25$	= 1,342.
$N(1-s) = 1{,}342 \div 22$	= 61.
Thrust	= 33.2.
$T_1 = \dfrac{33.2}{2.89 \times 4.84 \times .603}$	= 3.937.
$Q_1 = \dfrac{2\pi \times 115.76 \times .68}{2.89 \times 10.648 \times .603}$	= 2.666.
$(1-s) = 2.666 \div 3{,}945$	= .677.
$N = 61 \div .677$	= 90.1.
Speed of ship $= 13.25 \times 1.14$	= 15.1.
$Wp = 115.76 \times 2{,}240 \times 2\pi \times 90.1 \div 33{,}000$	= 4,448.
I.H.P. $= 2 \times 35 \times 4.5 \times 5{,}945 \times 90.1 \div 3{,}300$	= 5,112.
or $= 4{,}453 \div .87$	= 5,112.

It will be noted that with the assumption of constant torque, any change which will increase N will result in an increase in speed and *vice versa*, unless the conditions are such that the efficiency is seriously affected at the same time. This is readily seen from the fact that increase of N with constant Q will give increase in I.H.P. and, with efficiency substantially the same, this will give increase in speed. It may result, however, with constant decrease of propeller capacity and increase of revolutions that slip would finally so increase as to lead to a marked decrease in efficiency and a loss of effective power sufficient to balance or exceed the gain in I.H.P. In such case the speed would remain constant or decrease in accordance with the variation in the power actually effective.

The accuracy of the results obtained by the method of solution outlined for the problem as stated will depend, of course, on the accuracy with which the formulae employed are able to represent the operation of the propeller on the one hand, and the resistance of the ship on the other. In both directions much is left to be desired. The purpose of the present article is not, however, to discuss propeller or resistance formulae in themselves,

but rather to show in what manner the formulae or methods of computation at present available may be most conveniently arranged for the solution of the particular problem here proposed.

APPROXIMATE SOLUTION FOR EFFECTS OF SMALL CHANGES.

As already noted, the problem which we are here considering may arise in connection with trial results, where, with the assumption of a fixed steam pressure, it is desired to estimate the consequences of slight modifications in the propeller. Fixed steam pressure means constant turning moment, which is, of course, known, and the problem may then be treated along the lines already described. In such cases, however, it is possible to reach a close approximation to the consequences of slight changes by the use of a series of relations which we may develop in the following manner.

Let us suppose that the changes in the propeller are such as to result in an increase in the revolutions by a small fractional part, which we may denote by a. That is, if N_2 and N_1 are the revolutions in the two cases, then we assume that $N_2 \div N_1 = 1 + a$, in which a is a small fraction.

Then, since with fixed steam pressure the power will vary with the revolutions, we shall have an increase in power by the same fractional part a; and, since power will vary nearly as the cube of the speed, it follows that the speed will increase by the fractional part $a \div 3$. But speed is proportional to $p\,N(1-s)$. Now suppose that p has been increased by the fractional part b. Then in the expression $p\,N(1-s)$ we have p increased by the part b, N increased by the part a, and $p\,N(1-s)$ increased by the part $a \div 3$. Then it follows that $(1 - s)$ will be *increased* by the part $(a \div 3 - a - b) = (-2a \div 3 - b)$ or *decreased* by the part $(2a \div 3 + b)$. Now for usual values of the slip $(1 - s)$ will be not far from $3s$. Hence s will be increased nearly by the fractional part $(2a + 3b)$. Now examination shows that for ordinary values of the slip, increase in the slip by the part x will give an increase in the factor k by the part $.6x$. Hence in the present problem increase in slip by the part $(2a + 3b)$ will result in an increase in k by the part $(1.2a + 1.8b)$. Then, since N^2 will be

increased by the part $2a$, it follows that N^2k will be increased by the part $(3.2a + 1.8b)$.

We may now invert these relations and summarize them as follows:

Suppose

p increased by the part b, and $(d^2 p^3 l\, im)$ decreased by the part m, or $Q \div d^2 p^3 l\, im = N^2 k$ increased by the part m,

then

N will be increased by the part $(m - 1.8b) \div 3.2$;
power will be increased by the part $(m - 1.8b) \div 3.2$;
speed will be increased by the part $(m - 1.8b) \div 9.6$.

Thus, for illustration, suppose p left unchanged, d decreased by 2 per cent. of itself and the other factors so modified as to give a further decrease of 1 per cent. Then the total decrease of $d^2 p^3 l\, im$ will be about 5 per cent. Then $m = .05$, $b = 0$, and N will be increased approximately by 1.56 per cent., the power in the same ratio, and the speed by about .52 per cent.

Again, in the preceding problem, starting with the values taken and the solution reached, suppose the diameter to be increased by 3 inches, the pitch decreased by 12 inches, and the area decreased so that by estimate the factor im is decreased by 6 per cent. Then d will be increased by about 1.5 per cent and d^2 by about 3 per cent; p will be decreased by about 4.5 per cent. and p^3 by about 14 per cent; l will be increased by about 8 per cent., and im decreased by 6 per cent. The whole will result in a decrease in N^2k by about 9 per cent. Then $m = .09$ and $b = -.045$. Then $(m - 1.8b) = .09 + .081 = .171$, and the revolutions and I.H.P. will be increased by about .0534, or 5.34 per cent., and the speed by about 1.78 per cent. For the new revolutions we should therefore expect about 94.9, and for the speed 15.37. Where the changes are in both directions, increase and decrease, the algebraic signs + and —, will properly provide the necessary control, if used in the manner outlined above. While these relations are only approximate, they may often be made use of for the purpose of noting the general direction of the results of proposed changes, and of reaching at least an approximation to their values.

THE EDUCATION AND TRAINING OF NAVAL OFFICERS.

By Hayden Eames, Member.

It is almost a platitude to assert that in attempting to prescribe or suggest a scheme of education for application to a given organization, the exact end for which the organization has been created should be kept in mind and the educational scheme devised strictly to accomplish that end. The necessity for this sequence in the consideration of the subject is, however, often overlooked, and not infrequently the end to be attained is wholly crowded aside or obscured by meretricious or interested considerations. If this position is a correct one, the discussion of a scheme of education for naval officers necessarily requires the preliminary consideration of the very questions which many officers hoped would be forever solved by the "Personnel Bill."

Although undoubtedly of benefit as a harmonizer, unfortunately as a means of placing Naval Organization on a stable basis, this Bill was rendered abortive by the inevitable series of compromises in which logical considerations usually gave way to illusory ones based mainly on tradition.

Upon the questions involved in the late line and staff controversy I had, perhaps, the misfortune of entertaining views radically at variance with those generally held by either party.

It has always been my belief that the difference in the duties of the administrative and the technical branches should be more distinct than has ever been the case in the history of the Navy, and my convictions upon this point have been strengthened by relations with, and observation of the workings of both large and small industrial organizations.

In the course of fourteen years, it has been my good fortune to be able to include in the list of such organizations those in-

volving the extremes of relative distinction between administrative and technical duties.

Military and naval organizations, perhaps even more than others, suffer from adherence to tradition, and we all know and have often laughed at practices perpetuated long after the necessity for them has become obsolete. In the elimination of obsolete customs our naval organization has made tremendous advances in the last twelve years, but the traditions peculiar to the conditions of the old sailing-ship days are by no means eradicated. The so-called technical problems of those days were almost wholly empirical. The difference between a ship carpenter and a naval architect was then rather a difference of duties and opportunities than of education, or even ability. The step between a blacksmith or armorer and ordnance engineer was not a very wide one even up to the middle of the last century. On the part of any man of ordinary intelligence, therefore, there was really no obstacle to the acquirement of all the general and so-called scientific knowledge of the naval profession of that day to the utmost detail. The technical knowledge required for the relatively comprehensive and complete naval education of today would involve the mastery of the arts of two or three complete professions.

It is my belief that the administrative branch should be relieved of all strictly technical work and the scope of its duties limited to administration and operation; the latter being extended to the ship's motive power as well as to all machinery. The duties of the technical branch should be extended to cover the supervision of the construction, as well as the oversight of the maintenance of *all* the apparatus, including the ship's guns, motive power and other appliances. Except for purposes of education and experience, the employment of such trained engineers upon such duties as engine-room watch is a prostitution of the magnificent education which they are supposed to receive, and which I hope to see extended. Certainly, no industrial institution would receive anything but ridicule for the employment of highly-trained mechanical engineers and engine designers as stationary engineers in charge of only the operation of a power plant. Clash of authority on account of this proposed sharp

distinction of duties, in conjunction with the elimination from the technical branch of all administrative functions, can be easily prevented. The Navy need but to recourse to the common-sense methods frequently employed in industrial pursuits, such as the detail of men by the administrative branch on shipboard to act temporarily under the direct orders and supervision of officers of the technical branch in effecting repairs, or in the accomplishment of other duties connected with the proper maintenance of the equipment. We have, no doubt, often seen this identical course pursued in emergencies. I know of no reason why it is not equally satisfactory for routine purposes.

A radical difference in spirit exists between an administrative and a technical training. The former requires methods as concrete as the latter requires abstract ones. The world "marks the hits, but not the misses," and such conspicuous examples of administrative capacity as that of the retiring distinguished engineer-in-chief tend but to obscure the comparative incompatibility of the two kinds of training.

Daily and hourly evidence to the contrary notwithstanding, it seems to be a very common belief that any one of ordinary intelligence, whatsoever his training, is capable of the administration of almost any enterprise. Except in extraordinary cases of special aptitude, it seems to me preposterous to expect from a man without special training or experience a satisfactory performance in the erection of an organization; selection, discipline and development of the personnel composing it; the delegation of co-extensive power and responsibility without loss of personal control; the insurance of a steady adherence to the purpose for which the organization was created, without unduly restricting the development of individuality and individual methods; and the harmonizing to a common end of the conflicting requirements of several departments—for such are a few of the objects which the administrator of any considerable enterprise is required to achieve to a greater or less extent.

In naval administrative work a very important addition to the above is the necessity for that alertness, instant grasp of a situa-

tion and prompt decision hardly to be expected from those in the habitual practice of the abstract and contemplative methods required in high-class engineering.

It is a natural corollary of the above views that the technical education of the administrative and operative branch should be extremely thorough in all questions relating to the design and operation of naval equipment. In addition to this, that branch should have knowledge enough of questions of construction, and in general the problems with which the technical branch is confronted, to insure a proper co-operation between the two branches and a proper judgment on the part of the administrative branch as to the possibilities of construction and repair and the requirements of routine maintenance. It is the lack of an equivalent technical knowledge on the part of bankers and large investors that has been not only responsible for the puerile expectations of such people as to industrial performance, especially in enterprises of a novel technical character, but also largely for their gross misrepresentations to the public on the same subjects which have usually been unjustly ascribed to dishonesty.

I fancy no considerable objection, except possibly upon the ground of available time, will be raised against this extension of technical education in the administrative branch, but I would hardly expect the same indulgence for the proposal to transfer to the technical branch the construction and maintenance, say, of Ordnance.

The available length and scope of this paper hardly admits of the detailed discussion which this point may *seem* to require, but certainly does *not* merit. I can see no distinction between the manufacture of ordnance and that of any other mechanical contrivances, except such as lies in the well known mediaeval and highly meretricious considerations that made the manufacture of military weapons Spain's only industrial heritage from the Moors and included their manufacture as a part of the "art" of Benvenuto Cellini.

I shall never forget the consciousness of the waste of government time and money which I experienced in learning machine-shop practice when detailed as an Inspector of Ordnance, fully

conscious all the time that there already existed in the Navy a corps fundamentally trained in this identical field.

The only other technical branch which would seem to admit of special comment in an article of so general a character is that of Navigation, which, for practical naval purposes, being almost wholly applied to the "operation" of the naval equipment, should still remain as part of the training of the administrative branch only. Such a marked extension of technical education as herein proposed for both technical and administrative branches immediately suggests the necessity for the omission of some of the studies now pursued at the naval school. While I believe that the study of the English language is a most important and desirable one, I should have no fear of the result of dropping all such collateral branches of it as History and other sub-divisions of this branch tending solely to general culture. I do not recall among my naval acquaintances any one who really learned any history comprehendingly at the naval school, nor can I recall but very few officers who failed to fully qualify themselves on this and other similar subjects after graduation.

That the education of the two branches should start and continue for a certain period on identical lines I believe most desirable, if for no other reason than to insure the sympathy which tends to insure coöperation in after life. At what time the educations of the two branches should diverge; the extent, if any, to which the technical branch should be subdivided; the manner and extent of detail of members of the administrative branch in consultation upon questions of construction, and other similar precautions necessary to insure that the knowledge of the *whole* Navy rather than *a part* of it shall be available in such cases, are all questions which do not appear to me to be within the scope of this article, but none of which appear to be of difficult solution.

In the consideration of this subject it should be borne in mind that any scheme of organization or education for an organization must be contemplated with a view to application to the *average* man available. It would be utterly futile to devise an educational scheme which would attempt to provide in advance for the sudden assignment of especially suitable men in emergencies,

nor should accidental exceptions in the past and the present be allowed, in even a minor degree, to affect the fundamental outlines of such a plan.

The extraordinarily high morale of the naval service, and the ever readiness of its individuals to sacrifice their personal interests to those of their country and service, has always acted as a very effective palliative to the more or less illogical general and detail organization of the Navy, which, for the last twenty years, has resulted from the now happily gradually fading application of traditional principles to wholly changed conditions. It was the total failure of the majority of the American press to even remotely conceive of the nature of this morale which, during the Spanish war, resulted in their continual "discovery" of interested motives where such motives did not exist. I have every confidence that this same morale can be relied upon to eliminate the almost puerile considerations of self-interest and vanity which heretofore, in the case of very rare but somewhat powerful exceptions, have stood in the way of placing this whole question on a common-sense basis.

THE NEW ENGINEERING BUILDINGS, UNIVERSITY OF MICHIGAN.

By Mortimer E. Cooley, M. E., Professor of Mechanical Engineering, Member.

The world of today is prone to look upon Engineering as a modern science, and by many the profession is thought to be young and in its beginning. Relatively this may be true. Comparatively few, however, appreciate the part played by the engineer in the growth and development of the world—not only the world of today, but of that very early times. It is a noble profession with a noble ancestry, and has been an important factor in advancing civilization.

How many students of ancient history pause in their studies to consider what must have been the training leading to such achievements as the building of Pyramids and the Pharos of Egypt? The history of engineering progress is apart from the object of this paper; this contribution having been prepared for the purpose of disclosing to the busy men of other professions the modern trend towards engineering education and the extensive facilities that are being provided to equip our youth for engineering work.

The oldest engineering school in this country is West Point; founded in 1802. This was followed in 1835 by the Rensselaer Polytechnic Institute in Troy, New York, and in 1845 by Union College, Schenectady. In 1846 a course in engineering was founded in the Lawrence Scientific School at Harvard University, and in 1847 in the Sheffield Scientific School at Yale. The engineering department of the University of Michigan dates back to the year 1852, although in the act of the legislature of March 18, 1837, one of the sections mentions that there shall be established in the University a "professorship of civil engineering and architecture." The engineering department of the University of

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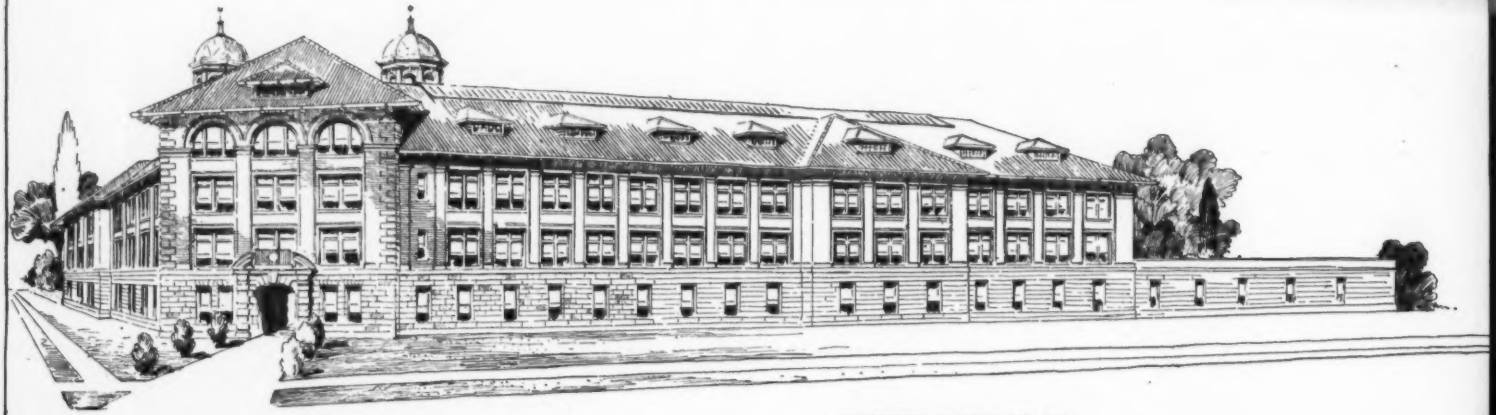

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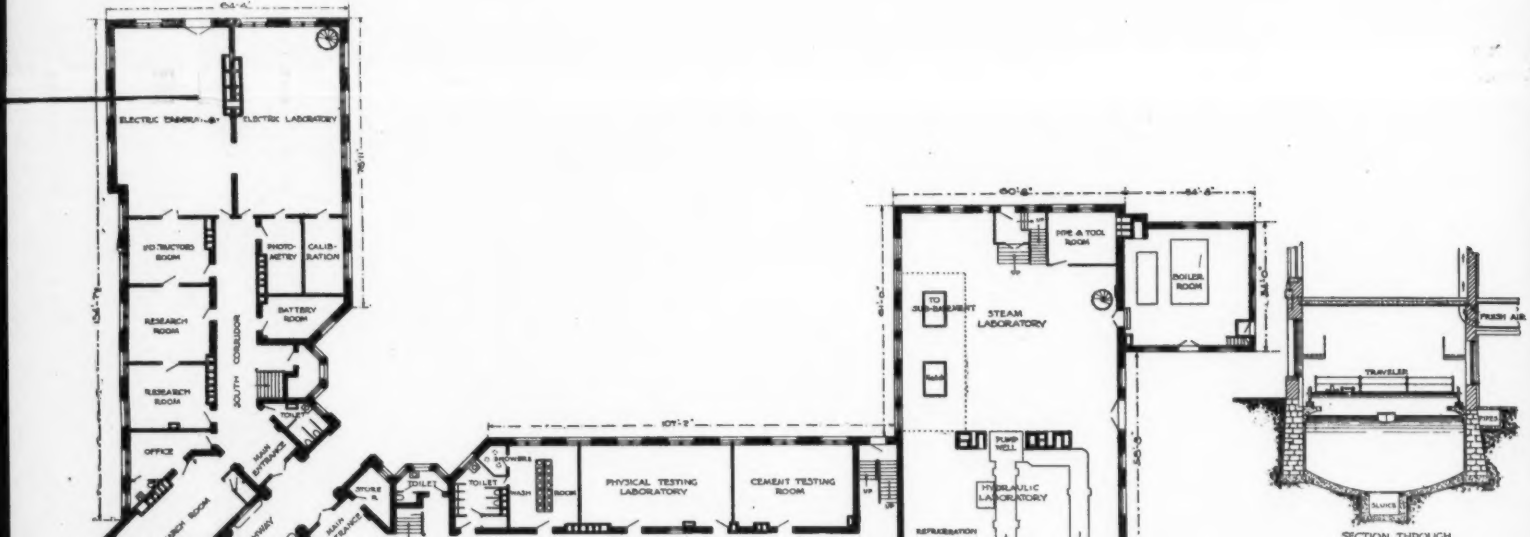
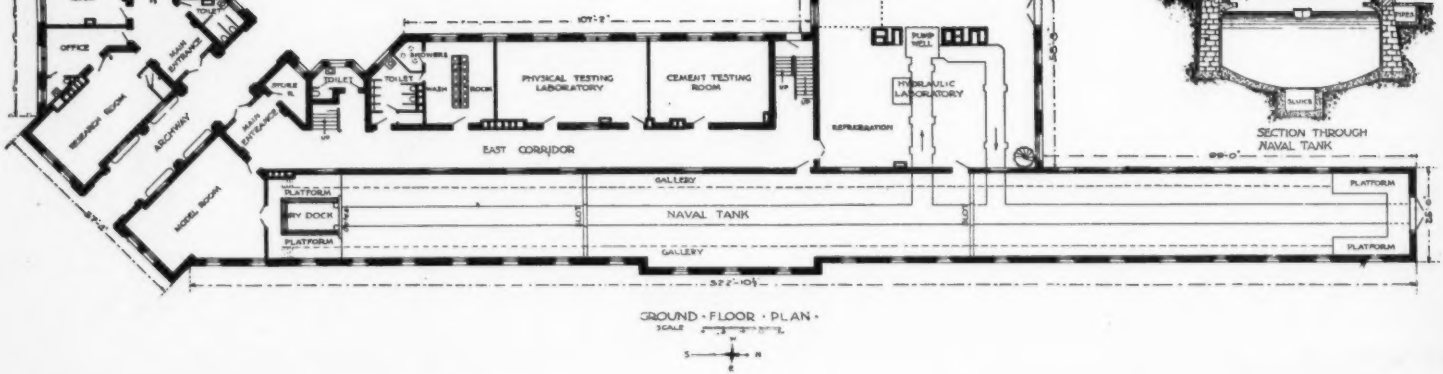
· UNIVERSITY · OF · MICHIGAN · ANN · ARBOR ·
· MASON & KAHN · ARCH'TS · DETROIT ·

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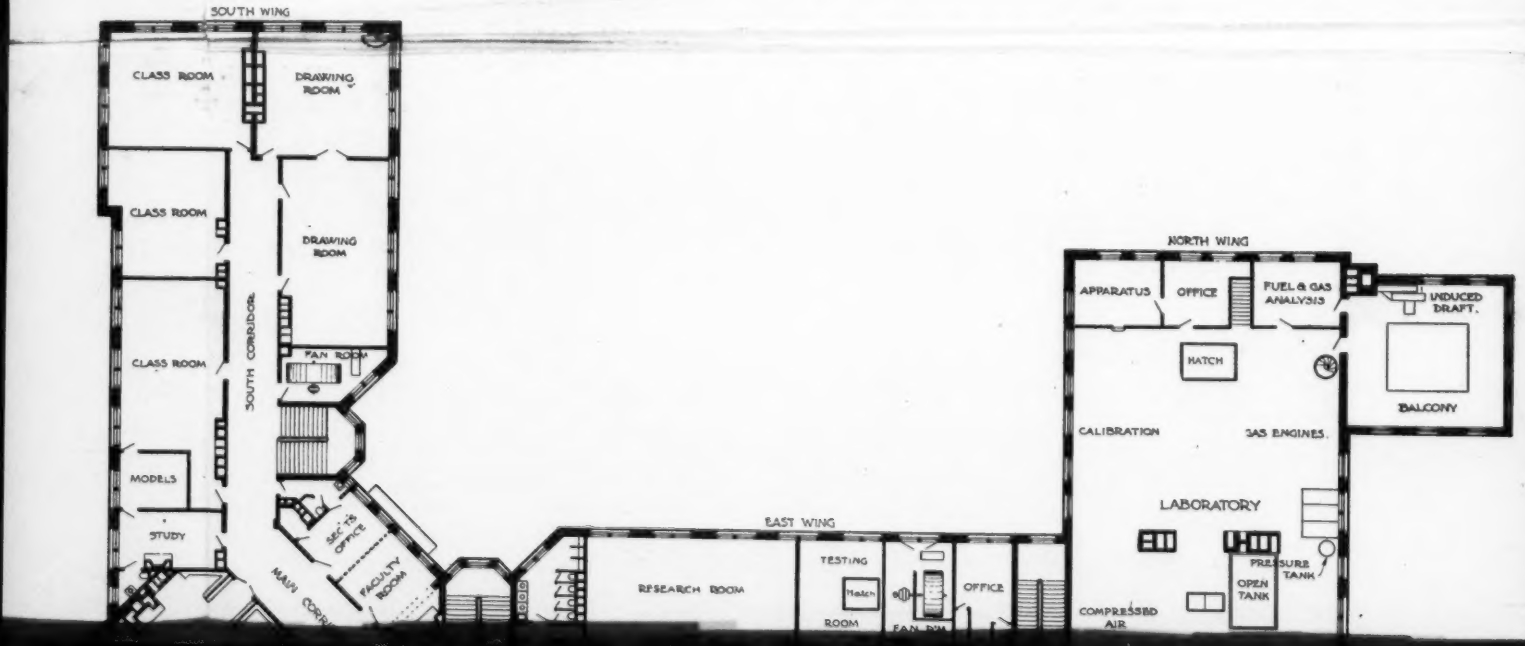
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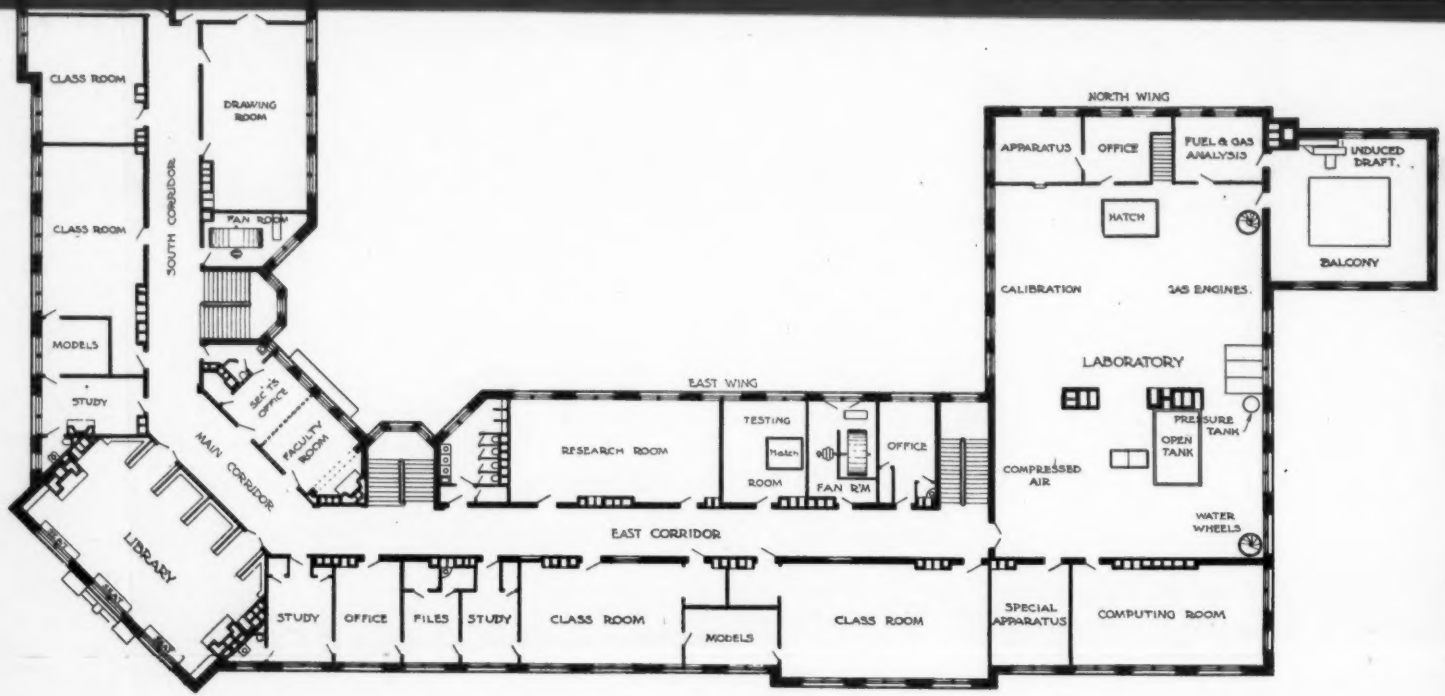

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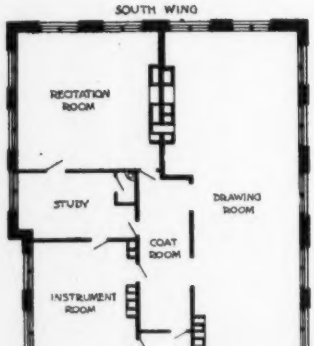

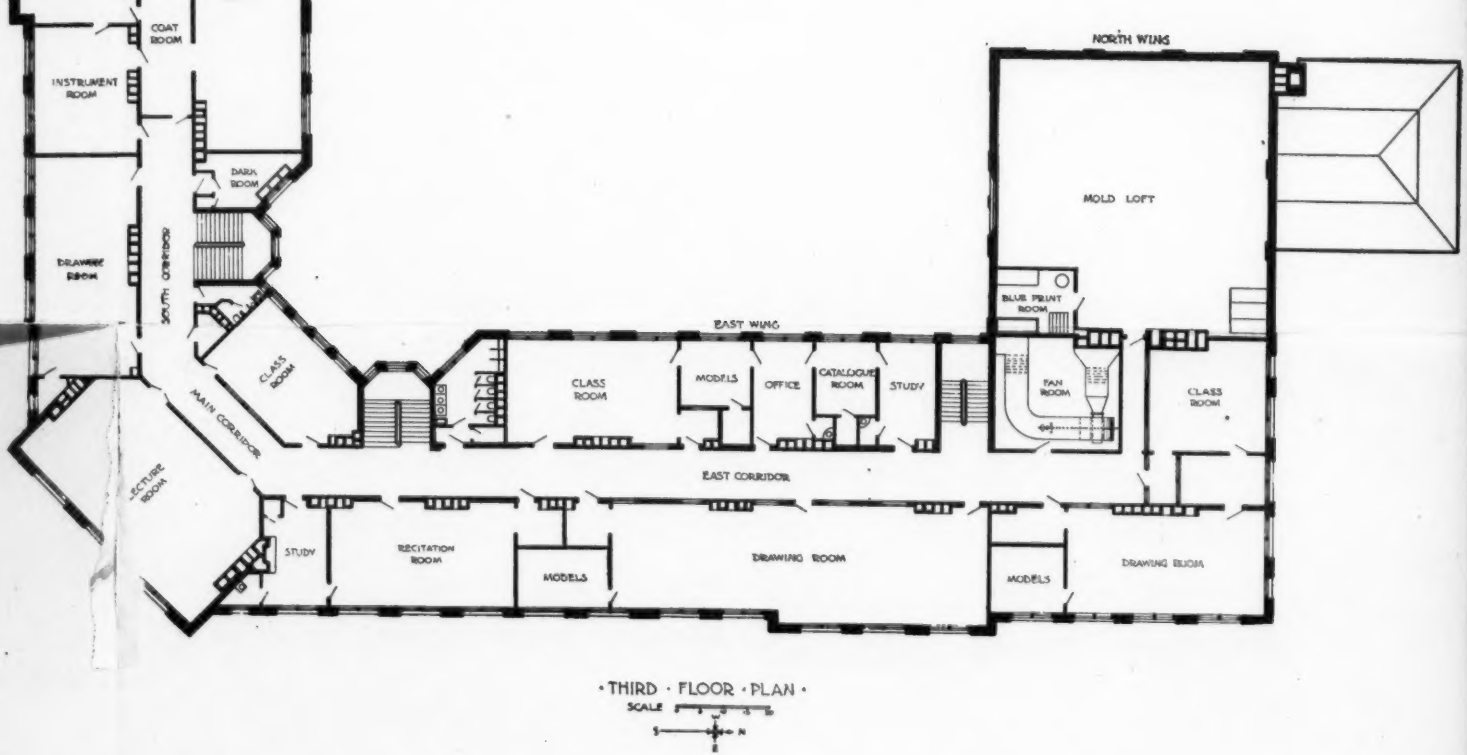

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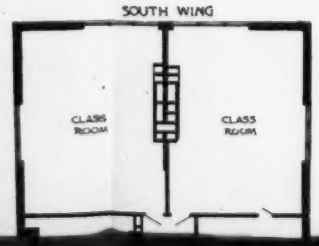

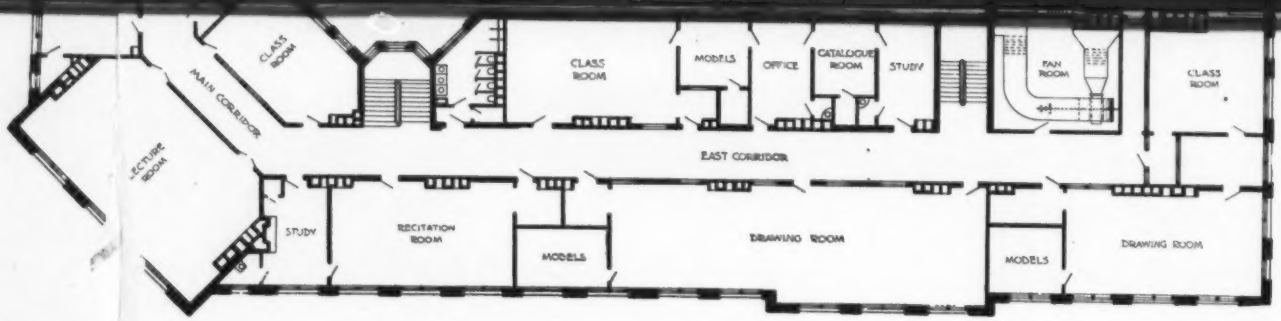

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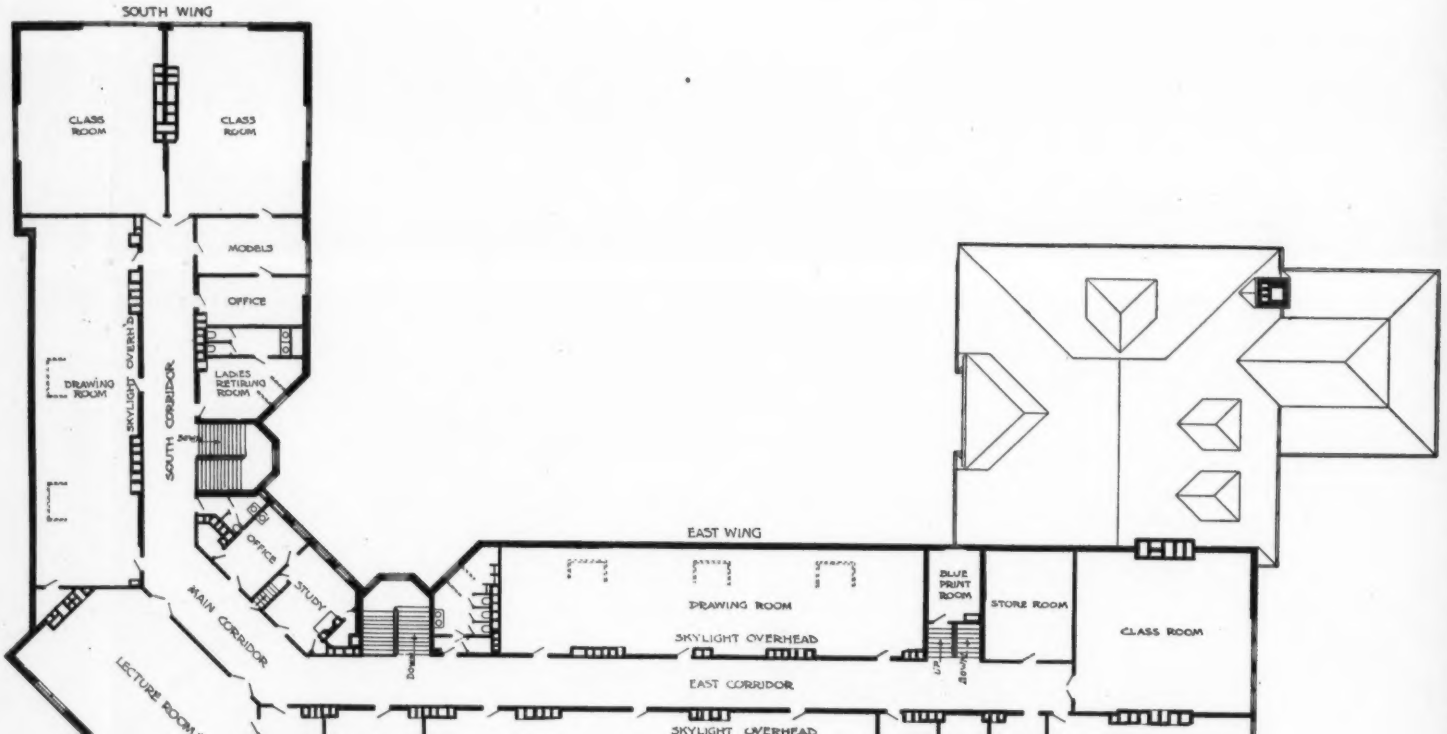

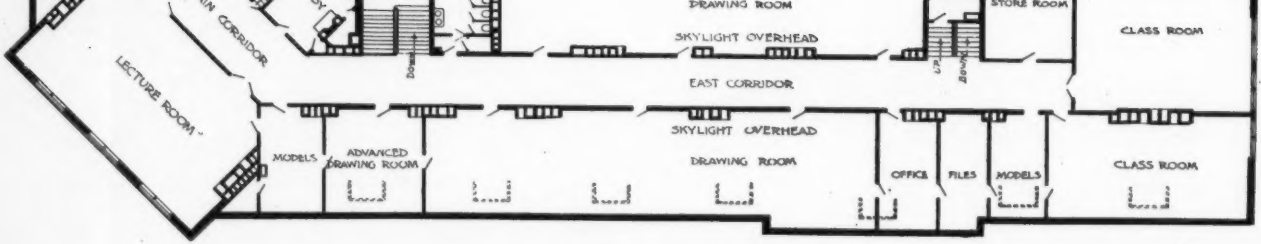

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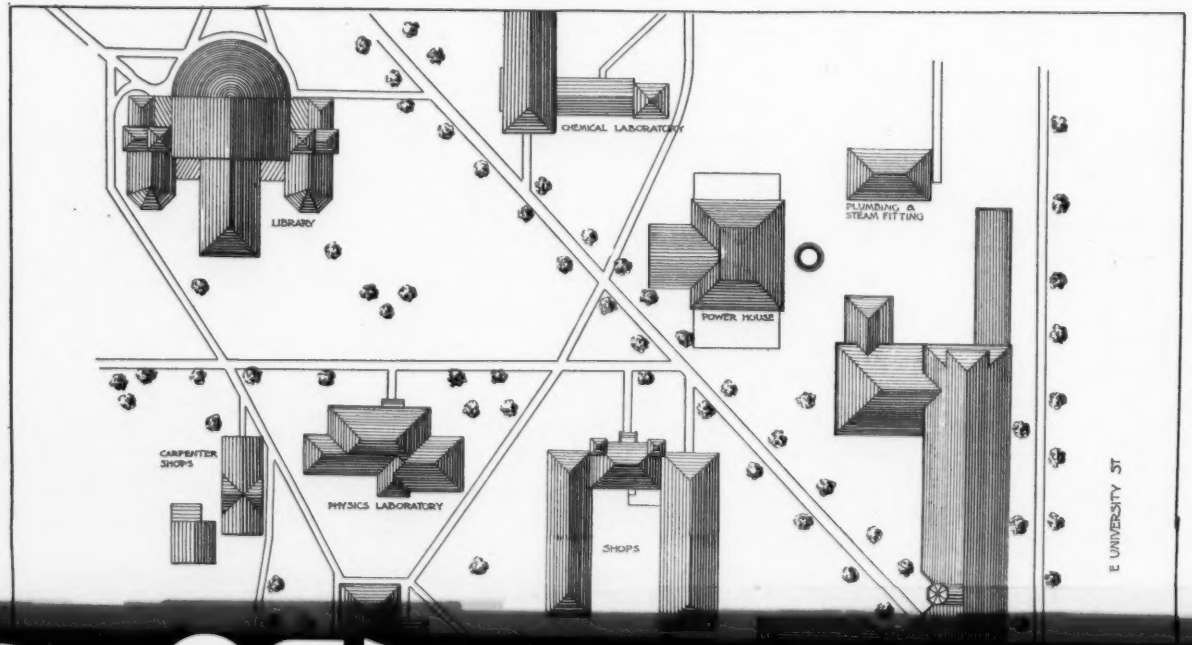

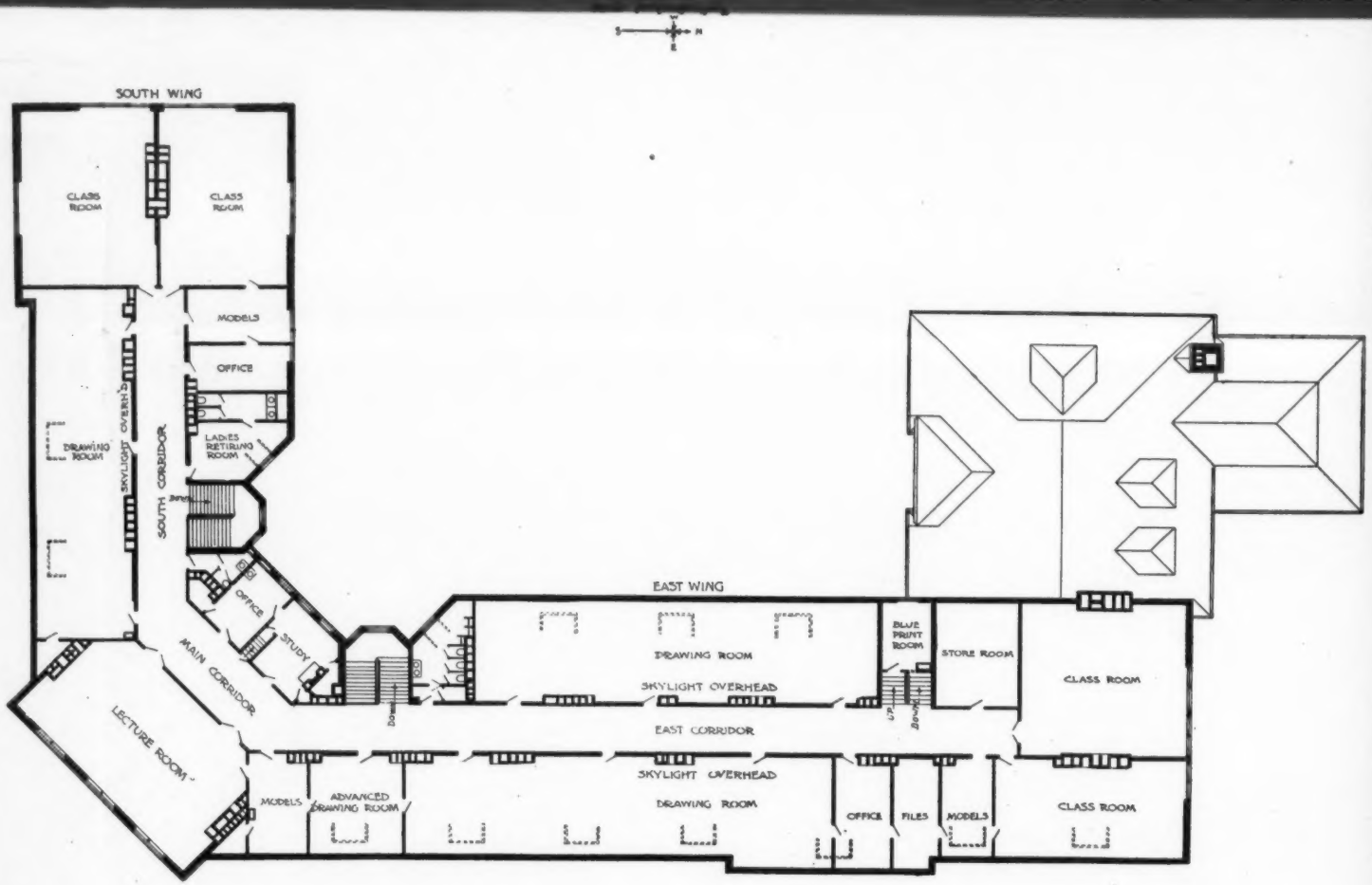

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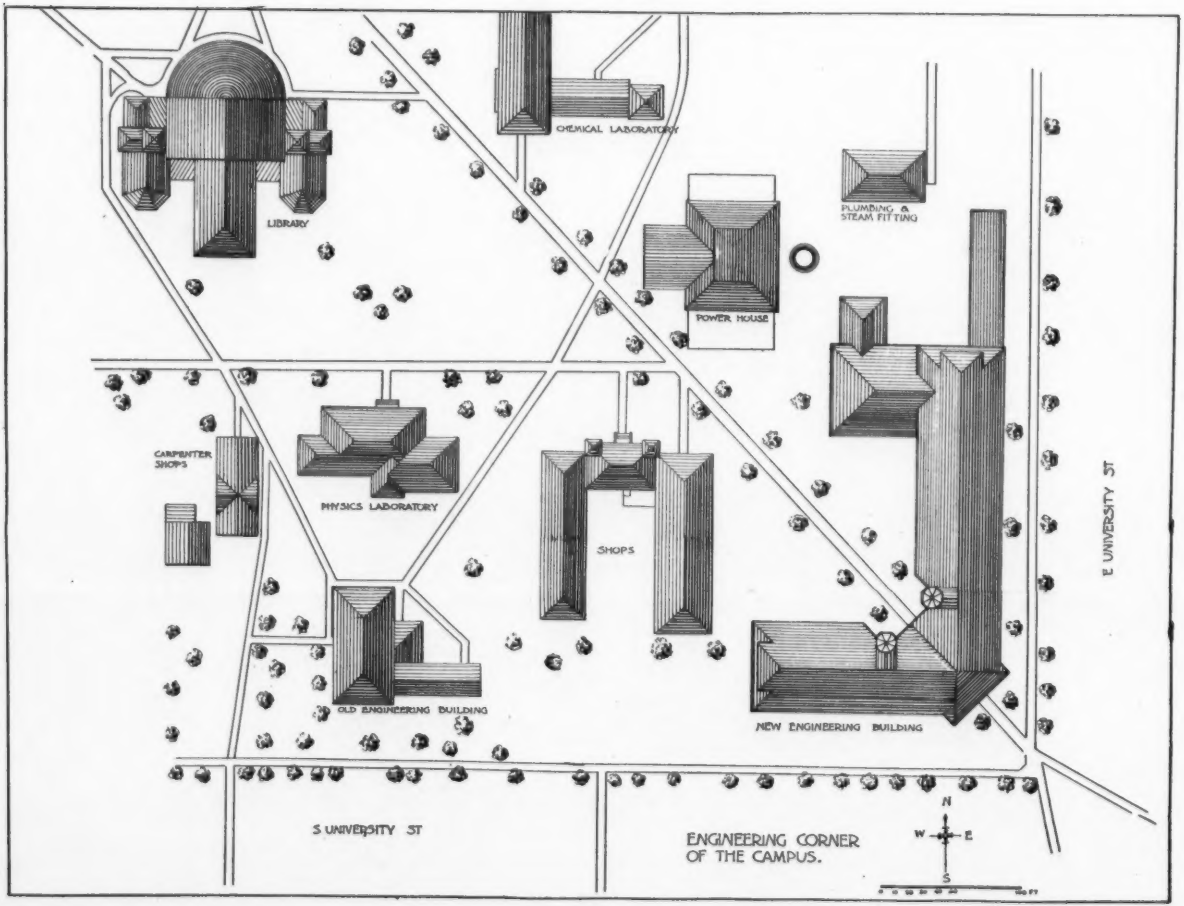

ENGINEERING CORNER OF THE CAMPUS.

Michigan may, therefore, be ranked as one of the oldest in the country.

During the past fifty years the half dozen engineering schools organized before that period has increased many fold, and today nearly every State in the Union numbers one or more technological institutions. These scientific schools are in many cases connected with the State University, and in a number of instances constitute the principal or leading department of the institution. In several of the commonwealths more than one engineering school is supported by State aid. In general, the support is very liberal, and increases from year to year as the legislative bodies awake to the importance of the work being done at the Colleges of Mechanic Arts.

The principal growth in our engineering schools has taken place during the past twenty years, and during the last ten years this growth has been phenomenal. Each of the branches of the profession has well defined curriculums of studies taught by trained experts, and the number of graduates annually turned out is very large.

Notwithstanding the fact that the graduating classes in the technological colleges are increasing very rapidly, there are not sufficient graduates to meet the demand. Numerous and pressing applications are flowing into all our engineering schools from manufacturing and transportation companies for technically trained men. It is significant that while ten years ago technical graduates were looked upon askant by many employers, especially in mechanical engineering, because of a lack of practical experience, today the employers say: "We do not ask for experienced men; all we want are men who have had a technical training; we will give them the experience ourselves."

For several years past, and again this year, a number of our larger industries have sent in requests for our graduates, saying that they can use the whole of the graduating class in certain lines of work. In some of our schools there are standing requests preferred by the very largest industries of the country for the brightest of our graduates. Such graduates are promised any advancement they may command and deserve by their energy and ability.

In explaining this phenomenal demand for technical graduates must we not associate it with the growth and development of our country; and may it not be true, as pointed out by a special correspondent of an English journal sent to this country to investigate our industrial condition, that the cause of our tremendous development is due to our strength in professionally trained men.

In 1880 the number of engineering students at the University of Michigan was less than fifty. At that time only civil engineering was taught, but now the curriculum embraces courses in civil, mechanical, electrical, chemical and marine engineering, as well as a special course in naval architecture. In 1890 the attendance in the engineering schools of this University was 159; in 1900 it had risen to about 300, and today there are over 600 students, with a prospective attendance in 1903–04 of 700 to 800. This rapid and continual growth makes clear the reason for the construction of an engineering building. Work on this building was begun in July, 1902, and it is expected to be ready for occupancy late in the year 1903–04. At present the walls are up, and the roof will be in place by September 1st.

In planning the building eighteen months ago some difficulty was experienced, owing to the congested condition of our campus, in selecting a design affording the accommodations desired by the different departments. As will be seen by reference to the drawing showing the engineering corner of the campus, there was very little space available except in the southeast corner, and this was cut by a prominent landscape feature known as the diagonal walk, which it was desired to preserve. At the same time it was absolutely necessary to find room for the proposed naval tank. Eventually the idea was conceived of spanning or arching the diagonal walk, and this plan gave the room required without sacrificing the beauty of the grounds or removing any buildings already erected.

The campus is square, one-fourth mile on a side, and contains 40 acres. The engineering corner is approximately one-quarter of the campus, the relative location and size of the different buildings being shown to scale.

The long stretch of ground on East Avenue suited admirably the general design and arrangement of the projected buildings. It permitted the installation of the proposed naval tank, a basin 300 feet in length. By extending the north wing sufficiently westward the steam engineering section approached conveniently near the main power house, and thus that building and its equipment can be easily visited during study hours by instructors and students. The end of the south wing is contiguous to the shops and foundry, and there is plenty of room for a front to be erected on the shop building facing South University Avenue in the future. When additional room becomes necessary the new building can be extended to the north over the naval tank, with an additional wing projecting to the west, the old anatomical laboratory, now given up to a shop for plumbing and steam fitting, being removed if necessary.

The relative location and size of the different buildings on the engineering corner of the campus are shown in the several plate.

The design of the building was entrusted to Mason & Kahn, architects, of Detroit. A general idea of their satisfactory solution of the architectural features of this somewhat difficult problem may be had from the accompanying sketch in perspective.

The building will be a handsome structure, by far the largest and the finest on the campus. The archway over the diagonal walk is unique, and the quadrangle with its beautiful trees produces a striking effect.

The lower story is of Bedford limestone in fairly large blocks, with deep reveals, presenting a solid and substantial appearance. Imposing stone arches project boldly from the building at the ends of the arcade, which is also of stone, with massive pilasters surmounted by groined arches. The main entrances are in this arcade. Small windows covered with heavy grille work, iron lanterns hanging from the keystones, and a tessellated floor must give the archway an imposing and beautiful effect, approached as it is by the sloping walk down through an avenue of maples.

Above the first story the walls are of a reddish-brown paving brick, with stone capitals at the head of the pilasters. The roof is to be covered with a red tile. The windows are to be of

plate glass, and are very large. The general effect should be thoroughly good, and creditable to the University. Some idea of the size of the building may be had from the ground-floor plan upon which the principal dimensions are given. The outer walls are over 1,100 feet around. The ground floor, exclusive of outside and inside walls, contains 25,800 square feet; the second floor, 23,300; the third floor, 23,200; and the fourth floor, 20,300; the total being 92,600 square feet.

On the ground floor the whole of the south wing will be devoted to laboratory work in electrical engineering. One of the larger rooms will contain direct-current, the other alternating-current machinery. This laboratory is designed to cover the different branches of applied electricity. The equipment will include the principal types of dynamos, motors, rotary converters and transformers, together with apparatus for testing. With the space available in the dynamo laboratories, it will be possible to introduce a change in the method of conducting the laboratory experimental work, so that several small sections of students may test and experiment on separate machines in different parts of the laboratory instead of only one large section being able to conduct such research as at present. By this change each student will secure much more instruction and experience than is now possible. One special feature will be a separate laboratory for telegraph and telephone engineering in which modern telegraph apparatus, including the quadruplex apparatus, and modern telephone exchanges, both manual and automatic, will be arranged for experimental work. It is expected to make experimental investigations on the newer methods now used in telephony and telegraphy. The laboratory will be well equipped for research work in the various fields of electrical engineering.

The east side of the east wing contains the naval tank, 22 feet wide, 300 feet long and 11 feet 6 inches deep, with 10 feet of water. A sluice extends along the bottom the entire length. The cross-section of the tank shows this sluice; also the curbs and the galleries at the sides. These curbs serve to deaden the waves and support steel rails resting on wooden stringers forming the track for an electrically-driven traveler spanning the tank.

The tank will be used in connection with the courses in naval architecture and marine engineering. It will afford means for a thorough investigation on various forms of ships. For this purpose models of from ten to twelve feet in length will be towed by the traveler, upon which is mounted a suitable dynamometer. The speed of the traveler can be varied to any desired extent, and the resistance of the model at the different speeds determined by the dynamometer. It is intended to carry out series of systematic investigations upon a great variety of ships' forms, so that the best form for any given set of conditions may be determined. Upon the traveler will also be mounted apparatus for testing propellers of various types and designs, both alone and in connection with a model vessel. Up to the present time most experiments of this character have been carried out in Government tanks, and a great part of the results obtained have necessarily been kept secret. Except in special cases, the results of tests conducted here would of course be available, and in time the ship-building industry of the country should profit thereby.

The models are prepared in the model room, adjoining the tank. It is proposed to use parafin wax for these, as this material is easily handled. The models are first cast roughly to the desired dimensions and afterward cut by special machinery to the desired shape. After being finished they are carefully measured, so as to obtain a trustworthy record of their exact form.

Numerous other experiments in connection with ships will also be conducted, such as tests showing the effect of bilge keels on the rolling and speed of ships.

The west side of the east wing contains the physical and cement-testing laboratories. Shower baths are provided next to the wash room for those engaged in specially dirty work, like boiler testing.

The north wing contains the hydraulic and steam laboratories, with a sub-basement under a part of the latter for convenience in collecting and weighing water from condensers above. The boiler-room extension from this wing to the north is designed for special work with high-pressure steam and superheating devices. A canal four feet wide and fourteen feet six inches deep

extends across the middle of the hydraulic laboratory. Water enters this canal from the sluice under the naval tank, and will be delivered by a centrifugal pump into a return canal leading back to the sluice. These canals are provided with slots to contain screens and bulkheads, in which orifices of various shapes and sizes may be formed. A dry pit is provided at one side of the canal and separated from it by a bulkhead. In this pit the conditions existing in the canal during a test can be observed. The centrifugal pump will have a capacity of about ten cubic feet a second. Two weighing tanks, each holding six hundred cubic feet, will also be provided for calibrating purposes. Rotary and piston pumps of at least one million gallons capacity are also contemplated. These, together with a pressure tank designed for 250 pounds pressure, will afford means for nozzle and motor tests. This pressure tank will extend through the second story. A mezzanine floor will be built above the weighing tanks in the north half of the hydraulic laboratory for additional machinery and apparatus.

In the steam laboratory, besides the usual machinery, it is designed to install a 300-horsepower vertical multiple-expansion engine with reheating receivers, all adapted for special investigations on the action of steam under pressure as high as 500 pounds per square inch, including superheated steam. Steam turbines are also contemplated.

Plans have been made for a refrigerating plant. This will serve for tests on mechanical refrigeration, and also provide means for maintaining low temperatures in cold rooms in the physical testing laboratories for investigating the effect of low temperatures on materials. A variety of experiments on somewhat new lines will be made possible by this plant. It is believed that very useful information may be thus obtained.

At each main entrance from the archway, at the north end of the east corridor and in the west end of the north wing, stairs lead to the second floor. The main stairs are of marble, the others of iron. The principal feature of the second floor is the library at the front of the central building, over the walk. This is a fine room, 30 by 53 feet, with paneled ceiling and walls and

arched alcoves for books. Two large fireplaces, window and alcove seats, and desks built into the walls, should make the room a cozy headquarters for the students between classes or while waiting to see the dean or the secretary, whose offices are across the corridor. The south side of the south wing, it is expected, will furnish class rooms in civil engineering, while those at the west end, together with the drawing room on the north side, will go to electrical engineering.

The east-side rooms of the east wing have been arranged for mechanical engineering, while those opposite are for laboratory or research work. The small testing room on this floor communicates with the testing room below by means of an open hatch through which the columns of a six hundred thousand pound testing machine rise from below.

The entire north wing on this floor will be devoted to laboratory work. This will include, among other things, gas engines, air compressors and water wheels. An open tank of seven hundred and fifty cubic feet capacity is supported near the ceiling and is supplied by the centrifugal pump. This tank will form the forebay for a variety of water wheels which will be arranged for demonstration and tests. The pressure tank, extending from the floor below, will also be available for experiments.

The pneumatic-air plant contemplates pressures as high as three thousand pounds per square inch, with both compressors and compressed-air engines arranged with special reference to securing reliable data on their performance. It is also hoped that facilities will be provided for determining the effectiveness of air at high pressure for transmission of power.

In the boiler room provision has been made for feed-water heaters of different kinds set up on the balcony; also for an induced-draft outfit, all being arranged for demonstration and tests.

The main ventilating fans are also situated on this floor, one in the south wing, the other in the east wing. These fans discharge into a large duct overhead in the ground-floor corridors. The fresh-air flues lead up the inside partitions or walls to the different rooms.

On the third floor the south wing is intended for civil engineering, design and drawing, and for surveying. In the east wing provision is made for mechanical engineering design and for mechanics. Other rooms will be available for class and lecture rooms, as named on the plan.

With the exception of one room, the entire third floor of the north wing will be given to naval architecture and marine engineering. The west end of this wing is a mold loft for the laying off of ships full size, as in shipyards. Here instruction will be given in the methods of fairing a ship's form, the preparation of of molds, templates, and other work usually done in a mold loft.

The fan on this floor has been specially arranged for investigating its operation under a great variety of conditions. It is provided with means for varying the speed, and for having any number of rows of pipes up to thirty-four on either the suction or the blast side of the wheel. If desired, all of the heater sections may be removed so as to run the fan with free discharge. This plant is a gift from the Buffalo Forge Company, as is also the induced-draft outfit. While adapted for experiments, this fan is also designed to ventilate and heat the north wing, the air being blown down to the rooms below.

The entire fourth floor is designed to accommodate the work in stereotomy, mechanism and drawing. There are, in addition to the lecture room in the central wing, four class rooms. These receive their principal light from windows. But the drawing rooms are lighted by a large skylight over the corridors, the arrangement being such that the light comes into the room in one direction only. These skylights show at the ridge on the sketch in perspective.

The space over the corridor between it and the skylight forms a large ventilating duct, corresponding to the fresh-air duct over the ground-floor corridor. All ventilating ducts discharge into this duct, which communicates with copper-covered ventilating towers at the head of the main stair wells.

Every room in the building is provided with flues for ventilation. This, as described, is a forced circulation, three fans being used for supplying the fresh air at a temperature of from 70 to

80 degrees. The north-wing fan, however, may use a higher temperature.

The toilet rooms are provided with suction fans placed in the ventilation towers. The building is to be heated by direct radiation. Both the direct radiation and the fan radiation are under thermostatic control.

Inasmuch as the class and lecture rooms will in general be used only in the mornings and the drawing rooms and the laboratories in the afternoon, the fresh air and the ventilating flues are all placed under pneumatic control from two central stations, so that rooms not in use can be cut out, and *vice versa.*

The plumbing is to be the best obtainable, and special attention has been given the problem of lighting. The building is fire proof throughout, the floors being of concrete.

The cost of the building, without heat, light, plumbing or ventilation, will fall just short of $150,000. It will probably not fall below $175,000 or $180,000 when ready for furniture and equipment. Such equipment as has been outlined would bring the cost to approximately $250,000.

As the volume of the building is 1,442,000 cubic feet, it will be seen that the cost of the building alone is approximately 10 cents per cubic foot; and when ready for occupancy, but without laboratory equipment, the cost will be under 12½ cents per cubic foot. In comparison with buildings erected elsewhere these costs are very low. This is due to lower labor rates, and to the fact that building materials are here readily available. The sand and gravel required for mortar and concrete was secured directly from the building excavation.

At the present rate of growth it will very soon be necessary to add to the building. As will be seen by the plat of the campus, this can be done very readily by removing the building now used for plumbing and steam fitting, and extending the east wing to the north over the extension of the naval tank. The walls of this tank extension are built heavy enough for this purpose. Another wing can be built extending to the west in the space now occupied by the plumbing and steam fitting building. There is room available for an east wing 400 feet long. This would add 100 feet to the naval tank, making it 400 feet long.

The present old engineering building will still be used as a recitation building for mathematics and modern languages in the engineering department. The room in the shops now used for laboratory work will serve for a very much-needed extension of shop facilities. This will provide a foundry 32 by 100 feet, a forge and structural iron shop, a machine shop, a woodworking and pattern shop, and a pattern loft, each 40 by 128 feet, besides three floors 35 by 50 feet for wash rooms, storage, drafting rooms and offices, giving a total floor area of shops of about 30,000 square feet.

The power house contains boilers of 2,000 horsepower capacity, including horizontal, fire-tubular and three types of water-tube boilers. It also contains two direct-connected generating sets of 75 kilowatts each with simple engines, and one of 200 kilowatts with compound engine. The exhaust steam is used for heating. A tunnel 6 feet 6 inches high and 5 feet 6 inches wide extends 3,000 feet around the campus, and carries all the steam pipes and electric mains to the different buildings.

All of this plant and a number of the heating systems in the different buildings have been planned for and are available for study and experimental work in engineering.

Such institutions as the University of Michigan, which possess equipment and facilities for instructing 800 engineering students, are substantially auxiliary military training schools, and contribute in no small degree to the military strength of the country. It ought not to be a difficult matter for the military and naval authorities to provide some means whereby the graduates of such institutions who have special aptitude and fitness for the military service could enter upon such work.

There are many officers in both Army and Navy who have been detailed to duty at such engineering institutions, and these men have a realization of the material that is available at such colleges for both military and naval needs. The technological schools are important auxiliaries to national defense, and the writer believes that a careful study of the work and equipment of the leading scientific institutions will convince thoughtful naval and military experts of this fact.

OBITUARY.

REAR-ADMIRAL DAVID SMITH, U. S. N. (RETIRED).

Rear-Admiral David Smith, who died at his home in Washington, D. C., on the 27th of May, 1903, was recognized as one of the ablest men of the Engineer Corps of the Navy. During his forty-four years of naval service he was identified with important events connected with its progressive development. His excellent professional record and his high moral worth and character were recognized both within and without the service, and at all times he upheld the honor and best traditions of the Navy.

He was born in Forfa-Shire, Scotland, December 13, 1834, and he was but a small boy when his parents moved to Massachusetts. He graduated from Phillips Academy, Andover, Massachusetts, in 1854. Possessing a constructive and engineering mind, and having a natural inclination for scientific work, he entered Harvard University (Scientific School) in 1854, and graduated from that institution second in his class in 1858.

Shortly after leaving Harvard University he began to give special consideration to the study of naval architecture and marine engineering. While visiting the Philadelphia Navy Yard in 1859 he met the Chief Engineer of the station, a man who afterwards became Engineer-in-Chief of the Navy, Commodore J. W. King.

As Mr. King had a great love, affection for and pride in the Engineer Corps of the Navy, he was exceedingly desirous that the Corps should be recruited in great part from the scientific graduates of the leading colleges, for he believed that with such material the Engineer Corps could eventually do better work as well as command more influence. It was the personal appeal of Chief Engineer King, combined with Mr. Smith's inclinations, that prompted the latter to enter the Navy. He passed first among the large class of candidates that were examined with him. The friendship of Engineer-in-Chief King extended through life, and there was hardly an important

engineering board organized during his incumbency as Chief of the Bureau of Steam Engineering of which Admiral Smith was not a member.

Between the years 1859 and 1867 Mr. Smith served on board the *Narragansett*, *Pocahontas*, *Lancaster*, *Saginaw* and *Tallapoosa*. On three of those ships he passed through a siege of yellow fever. While on board the *Tallapoosa* that ship encountered a gale which carried away both rudders, and he was very prominent in making the repairs which brought her into port.

In 1867 he was ordered to the U. S. S. *Wampanoag*, a vessel which was the fastest ship afloat at that time, and whose ocean speed was not exceeded for fifteen years afterwards. Extensive dock and sea trials were made with this ship, and Chief Engineer Smith was placed in charge of the engineers' department over two of his seniors who were detailed for duty in connection with the trials of that vessel.

In 1869 he was ordered as Assistant to the Chief of the Bureau of Steam Engineering. While doing this duty he was called upon to report as to the value and mechanical worth of various appliances that were submitted by inventors for use in the naval service. During this year he was highly commended by both the Secretary of the Navy and Judge Davis, who represented the United States at the Geneva Conference, for the clear and concise reports submitted by him in regard to the character of the repairs made to the Confederate cruiser *Shenandoah* at Port Phillip, Australia.

In 1875 he was made a member of the United States Steel Board, a commission organized for testing iron, steel and other metals. The results obtained by these tests helped materially to remove the public prejudice against the use of American tool steel, and it was the remarkable power of this officer in presenting a case that brought out so clearly the work done by the Steel Board.

In 1876 he was twice offered by the University of the State of Pennsylvania the Chair of Dynamic Engineering. He was a member of the McCawley and Stevens Boards on the *Puritan*, his duty there being of a very arduous nature, and one requiring

special knowledge of hull and engine construction and computation.

Probably the most important work that he ever rendered was his service on a Board for the better ventilation of ships. He was practically the originator of the system of exhausting foul air from each compartment of the ship, although the fans were so installed that they could be made reversible in case of need. The plan proposed by that Board, and the details of which were worked out by this officer, are practically used in all the large ships of the mercantile marine and navies of all nations.

During the past fifteen years he was made a member of many important Boards. His work in standardizing indicators and instruments of precision, as well as his power of conducting original investigation and research, made his services of great value to the Engineer-in-Chief of the Navy. He was a member of the Board appointed to reorganize the Civil Establishment of the Navy Yards. His engineering, business and executive ability admirably fitted him for this special work, and many of the most important recommendations were suggested by him. During his naval career he made a special study of the subject of refrigeration, for he believed that this question had a very important bearing upon naval efficiency. In 1893 he was ordered to the U. S. S. *Philadelphia*, as fleet engineer of the Pacific Squadron. In 1896 he was elected President of the American Society of Naval Engineers, and during his administration of affairs there was a noticeable increase both in prestige and influence of the organization.

On February 5th, 1903, by special act of Congress, he was advanced one grade in rank (that of rear-admiral). The bill for giving him this rank was carefully considered by both Naval Committees, and thus before his death he had the satisfaction of knowing that modest and faithful service had been recognized by the Government. It was ever his belief that the consciousness of duty well done was the greatest reward and honor that could come to any officer who, throughout his naval career, had regarded efficiency, integrity and loyalty as the highest attributes of those holding commissions in the naval service.

ASSOCIATION NOTES.

At a meeting of the Council of the American Society of Naval Engineers held July 10th, the resignation of Lieutenant C. W. Dyson as a member of the Council was presented and accepted, this officer having received preliminary orders to sea. Captain J. W. Collins, Engineer-in-Chief of the Revenue Marine Cutter Service, was elected to fill the vacancy thus created.

The Secretary-Treasurer was directed to call the attention of the members of the Society to Article XIX of the By-laws, which provides that "The annual subscription of members and associates shall be $5.00, payable in advance." In accordance with this action of the Council the members, associates and subscribers who are in arrears are requested to forward such remittance at the earliest practicable time.

The Council of the Society would inform the members that during the past year there has not only been a progressive gain in membership, but that increasing interest in the purpose of the organization is being manifested by many persons within and without the naval service.

The Council urges the members to point out to the Junior Officers of the Navy the advantages of membership in this organization.